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THE ANNUAL MEETING

The Annual Meeting this year will comprise sessions upon a greater variety of topics than have ever been arranged for a meeting. No less than ten sessions are contemplated with papers grouped to form symposiums upon various subjects. This broadening of the Society's work is the result of the sub-committees lately appointed under the direction of the Committee on Meetings. The publication of the papers for the Annual Meeting began in the September number of The Journal, and others will appear in the December number or be printed in pamphlet form for distribution at the meeting and published in later issues of The Journal. The papers already published are as follows:

The Principles of Valuing Property, Henry K. Rowell

Increase of Bore of High-Speed Wheels by Centrifugal Stresses, Sanford A. Moss

Investigation of Efficiency of Worm Gearing for Automobile Transmission, Wm. H. Kenerson

Measurement of Air in Fan Work, Charles H. Treat

The Baltimore Sewerage Pump Valve, A. F. Nagle

The Vauclain Drill, A. C. Vauclain and Henry V. Wille

Tests of a 1000-H.P., 24-Tubes High B. & W. Boiler, B. N. Bump

The V-Notch Weir Method of Measurement, D. Robert Yarnall

Dimensions of Boiler Chimneys for Crude Oil, C. R. Weymouth

Air in Surface Condensation, George A. Orrok

The Centrifugal Blower for High Pressures, Henry F. Schmidt

Measurement of Natural Gas, Thos. R. Weymouth

One whole session will be devoted to reports of technical committees which it is expected will be of as great interest as any other single feature of the professional sessions. One of the reports is an exhaustive document prepared by the Power Tests Committee. When finally revised and submitted to the Council, this will supersede the earlier reports upon methods for conducting trials of pumping engines, locomotives, steam boilers and steam engines, besides taking up the testing of other apparatus, such as gas and oil engines, water-wheels, and compressors, blowers and fans. The latest Society report upon any of these subjects was in 1902 and the earliest in 1890, twenty-two years ago, so that the need for revision is evident.

The general arrangement of the sessions will be substantially as follows:

TENTATIVE PROGRAM

Tuesday Evening, December 3

President's Address followed by the reception to the president-elect, the retiring president, Dr. Alex. C. Humphreys, and their ladies.

Wednesday Morning, December 4

Business meeting, followed by simultaneous sessions, under the direction of the Gas Power Section and the sub-committees on Machine Shop Practice and Textiles.

Wednesday Afternoon

Reports of Committees, Power Tests, Symbols, etc.

Wednesday Evening

Dinner to Professor John E. Sweet, Honorary Member and Past President, in celebration of his eightieth birthday and in recognition of his services to the engineering profession, given by his friends and members of the Society.

Thursday Morning, December 5

Simultaneous sessions under the direction of the sub-committees on Railroads, Iron and Steel, and Cement Manufacture.

Thursday Afternoon

Sessions on the Power Plant, and on Hydraulic and Pneumatic Apparatus, including centrifugal pumps, blowers and the measurement of flow of fluids.

Thursday Evening

Reunion and dance, with collation.

Friday Morning, December 6

Session of the Sub-Committee on Administration, presenting a review of the present state of the art of management.

The list of papers for the Machine Shop session is a strong one and includes the results of tests and investigations as well as machine

shop operations. At the Textile session the valuation of property and the special requirements of power plants for textile mills are the subjects for discussion. The sub-committee on Railroads have papers on train lighting, electric locomotives and the locomotive problem, and at the Iron and Steel session the most advanced achievements in iron and steel manufacture will be the subject. At the Cement session the subjects of depreciation and obsolescence and of accidents are to be treated.

On Friday the whole session, and the whole day if necessary, will be devoted to a review of the present state of the art of industrial management.

RAILROAD TRANSPORTATION

Arrangements for hotel, transportation and Pullman car accommodations should be made personally.

For members and guests attending the Annual Meeting in New York, December 3–6, 1912, the special rate of a fare and three-fifths for the round trip, on the certificate plan, is granted when the regular fare is 75 cents and upwards, from territory specified below.

a Buy your ticket at full fare for the going journey, between November 29 and December 5 inclusive, and get a certificate, *not a receipt*, securing these at least half an hour before the departure of the train.

b Certificates are not kept at all stations. If your station agent has not certificates and through tickets, he will tell you the nearest station where they can be obtained. Buy a local ticket to that point and there get your certificate and through ticket.

c On arrival at the meeting, present your certificate to the registration desk at the Headquarters. A fee of 25 cents will be collected for each certificate validated. No certificate can be validated after December 6.

d An agent of the Trunk Line Association will validate certificates, December 4, 5, 6. No refund of fare will be made on account of failure to have certificate validated.

e One-hundred certificates and round trip tickets must be presented for validation before the plan is operative. This makes it important to show the return portion of your round trip ticket at Headquarters.

f If certificate is validated, a return ticket to destination can be purchased, up to December 10, on the same route over which the purchaser came, at three-fifths the rate.

This special rate is granted only for the Trunk Line Association, including:

All of New York east of a line running from Buffalo to Salamanca, all of Pennsylvania east of the Ohio River, all of New Jersey, Delaware and Maryland; also that portion of West Virginia and Virginia north of a line running through Huntington, Charleston, White Sulphur Springs, Charlottesville, and Washington, D. C.

THE JOINT MEETING IN LEIPZIG

General plans for the itinerary of the Society's visit to the German Verein have been presented to the Committee on Arrangements by Dr. Matschoss, their special representative, and their adoption now depends mainly on an approximately correct list of members and guests who will compose the party.

Preliminary estimates of the cost of the excursion here given will, it is hoped, enable members to reply more definitely to our circular of August 27, for the cost of steamship and railroad fares depends directly on the number of participants.

We can accommodate from 160 to 200 first-class passengers; about 80 on two promenade decks, the others on the upper and main decks. Prices, according to location, range from $90 to $130 per berth. This includes first-class passage from New York to the German port, with meals and hot and cold salt water baths, and railroad fare to Bremen or Hamburg. The railroad fare from there to Leipzig will be $5 each on regular express trains, but if at least 160 members and guests take part, a special train can be furnished at about $2.50 per person. The excursion from Leipzig to Dresden, Berlin, Düsseldorf, Essen, Cologne, Wiesbaden, Frankfurt, Heidelberg, Nürnberg and Munich will cost from $23 to $26 each, with probably a similar liberal deduction if a special train is chartered for not less than 160 passengers.

German railways have first, second, third and fourth-class carriages. There is no difference between first and second-class except the color of the upholstery and the seating capacity, first-class having six seats and second-class eight seats per section. As a rule, well-to-do Germans travel in second-class carriages, first-class being patronized by high officials and foreigners unfamiliar with the customs of the country. For short trips of less than five hours, most

Germans on their summer vacation tours travel third-class, the carriages of which are roomy and clean, with plain wooden seats.

The figures above given are for second-class coaches throughout. Dining cars accommodate first and second-class passengers only. Hotel accommodations can be had at American prices if American style and luxury are desired. But the hotels patronized by native travelers are much cheaper, serve excellent meals and are scrupulously neat. Single rooms with private bath cost $2.50 to $3.50 per day; double rooms from $3.50 to $6.00. Single rooms without bath from $1.00 to $2.00; double rooms from $2.50 to $5.00, and excellent baths can be had on the same floor for 35 to 50 cents. In every city there are first-class swimming baths.

Good meals on dining cars and excellent meals in hotels range from 35 cents for breakfast to 75 cents and one dollar for dinner. Germans generally dine at midday, and take coffee or tea in public gardens where a band plays, and supper in garden restaurants; rates about 25 per cent less than in the United States.

In the matter of meals there is a radical difference between hotels in Europe and America. The steam table which reduces all viands to mediocre similarity is unknown in Europe. Meals are served at stated hours fresh from the chef's hands, each course distinguishable without reference to the bill of fare, incidentally reducing the cost of service. The result is that meals in the smaller and cheaper hotels are much better than in houses of the same class here.

With the very reliable Baedeker in your pocket you can travel on the continent at much less cost than here.

A gentleman who recently traveled in Germany and France with two ladies for three and one-half months reports total expense, exclusive of steamer fare, at $5.10 per day per person.

COMMITTEE ON ARRANGEMENTS

AMENDMENT TO BY-LAW 18

At a meeting of the Council, October 8, By-Law 18 was amended to read as follows:

B 18 The Council at any meeting may, in its discretion, permanently remit the dues of any Member of the Society who has paid thirty years' dues or who shall have reached the age of seventy years after having paid twenty-five years' dues, provided that notice of such proposed action shall have concurred in recommending that this action be taken. The Council may, in its discretion, restore to membership any person dropped from the rolls for non-payment of dues, or otherwise, upon such terms and conditions as it may at the time deem best for the interests of the Society.

MEETINGS OF STUDENT BRANCHES

With the reopening of the universities, meetings of the Student Branches of the Society, which now number twenty-five with a total membership of 700, have been resumed with customary vigor. Although it is possible to publish no more than very brief accounts of these meetings from time to time, much work of real professional merit is being accomplished in these branches, which deserves wider mention. In some of the branches, outside speakers of prominence in the engineering world are secured to address the college at intervals on vital aspects of the profession, while in others members meet to discuss the papers presented at meetings of the Society. Original papers are also secured and read at many of the branch meetings, and at least one of the colleges held last year a small convention, with afternoon and evening sessions and a carefully arranged program.

The Armour Student Branch opened its meetings for the season on September 19, with its annual smoker at which Prof. G. F. Gebhardt, Prof. A. J. Frith and Prof. C. E. Paul, Members of the Society, were the speakers. On October 9 a joint meeting was held with the student branch of the American Institute of Electrical Engineers, C. K. Baldwin, Mem.Am.Soc.M.E., giving an illustrated lecture on Belt Conveyors.

At the first meeting of the Ohio State University student branch, Prof. Wm. T. Magruder, Mem.Am.Soc.M.E., gave an interesting talk on the Society, its growth and what it stands for, with special reference to the value of affiliation with it.

A well-attended meeting was held at the Polytechnic Institute of Brooklyn on the evening of October 5, for the purpose of laying out plans for the coming season, for which the prospects are excellent. B. L. Heustis was elected president for the year, and A. Bielek, secretary.

The Student Branch of the State University of Kentucky elected the following officers at a recent meeting: R. R. Taliaferro, chairman, R. E. Mattingly, vice-chairman, F. J. Forsyth, secretary, and D. W. Perry, treasurer.

At the University of California, J. F. Ball was elected to the office of chairman at a meeting on September 14, and Jesse Blair to the office of vice-chairman.

The University of Missouri Student Branch held a meeting on September 30, at which Prof. H. W. Hibbard, Mem.Am.Soc.M.E.,

gave a talk on the value of affiliation. The following officers were elected: F. I. Kemp, president, R. Runge, secretary-treasurer, J. H. Pound, corresponding secretary.

Meetings were resumed by the University of Nebraska Student Branch on October 1, at which short talks on affiliation were given by Prof. J. D. Hoffman and Professor Raber.

The University of Wisconsin Student Branch held their election of officers on October 17, with the following result: W. A. Moyer, president, A. G. Peter, vice-president, W. K. Fitch, secretary, W. H. Schleck, treasurer. Committees were appointed by the president and the work for the year planned.

At Yale University a meeting of the Student Branch was held on October 11, at which Prof. L. P. Breckenridge, Mem.Am.Soc.M.E., was the principal speaker. Plans for the year were outlined by the president, C. E. Booth, and an interesting demonstration was given of a new combined stereopticon and reflectoscope, just installed at the Sheffield Scientific School.

DR. VON MILLER ELECTED TO HONORARY MEMBERSHIP

At a meeting of the Council, October 8, 1912, Dr. Oscar von Miller, director of the German Museum in Munich, was elected to honorary membership in the Society. Dr. von Miller who, in addition to his connection with the German Museum, is this year president of the Verein deutscher Ingenieure and a civil engineer of the city of Munich, has been prominent for a number of years in his profession. He was the designer of the electric central station in Kassel, the first central station in Germany with a converter from alternating to direct current, and also the central station in Heilbroün, the first alternating-current installation in Germany. He also designed and directed the construction of the electric trunk line Mecklenbeuren-Tettnang and the central stations at Nürnberg, Strassburg, Wiesbaden, Bozen-Meran, Riga, and others. He is the author of a book on the Supply of Electricity in Cities, and has received decorations from the governments of Italy, Sweden, Bavaria, the Grand Duchy of Baden, and from Württemberg.

Dr. von Miller was one of the founders of the German Museum of Masterpieces of Natural Sciences and Technology, of which he is now the director.

NEW SECTIONS OF THE SOCIETY

In addition to the meetings of the Society which are now held in New York, Boston, Philadelphia, St. Louis, San Francisco, and New Haven, committees have been organized in Cincinnati and Chicago in charge of meetings in those cities, with the following personnel: In Cincinnati, A. L. DeLeeuw, chairman, J. T. Faig, secretary, W. G. Franz, G. W. Galbraith, L. H. Thullen; in Chicago, Paul P. Bird, H. A. Bogardus, G. F. Gebhardt and A. L. Rice.

On October 17, the first meeting of the Cincinnati Section was held, jointly with the Engineers Club of that city. A. L. DeLeeuw, chairman of the section, read a paper on The Trend of Modern Engineering, which treated of the gradual change of the scope of the engineer from an inventor to a true economist, who sought to get the greatest amount of product with the least amount of effort or expenditure. The latest phase of this subject was especially considered, namely, scientific management, and how the task and bonus plan in connection with instruction sheets leads to economies, not only by its direct effect on the workingman, but also because it brings to the attention of the management every point of the equipment not up to its highest efficiency. The paper further discussed the effects of this system on the relation between employer and employee and how it establishes a well-defined contract between them, producing a beneficial effect and becoming in itself a source of economy.

DINNER TO DR. MATSCHOSS

At the conclusion of the Council meeting on October 8, an informal dinner was tendered by the members of the Council to Dr. Conrad Matschoss, Dozent of the Technische Hochschule of Berlin, an official representative of the Verein deutscher Ingenieure.

In Dr. Matschoss's remarks he spoke of his success in his tour of the United States in interesting men of affairs and engineers in the writing of their biographies and histories of the technical arts, so that there may be available in the future a record of the development of each of the industries. He remarked that the conditions in particular ages may be obtained from literature, but that no convenient report of the progress made in any art is at present obtainable, with a few exceptions. Dr. Matschoss also spoke of the increasing interest which every young man ought to have in his

profession, and the importance of developing this respect and zeal for one's work.

Mr. Swasey, who took the chair after Dr. Humphreys was called away, pledged the company to take up the work as outlined by Dr. Matschoss, and it is the hope of all that in the near future several of our honorary members and others prominent in the engineering profession will make good their promise to write autobiographical notes which the Society may publish uniformly with the edition of the Life of John Fritz, brought out under the auspices of the Society.

The meeting concluded with an expression of appreciation to the Verein deutscher Ingenieure of their invitation to the Society to meet with them in Germany next year and to make a tour of the industrial centers.

REPORT OF COMMITTEE ON PATENT LAWS

The following report on the patent law situation has been made by the committee appointed by the Council to investigate this subject, in which every member is directly or indirectly interested. The committee attended the patent conference held last spring in Washington between the bar associations and the engineering societies of the country, and have also been requested by the Council to continue to keep in touch with the situation and to report to them from time to time regarding recent developments.

To the Council:

Pursuant to our appointment on April 9, 1912, by the Council as a Committee on Patent Laws, we have continued to study the patent situation in Washington and the course of legislation which has been introduced in the House on the subject of Patents, amounting to more than 50 bills, most of which appear to be not in the interest of inventors and patentees.

In the early part of May, the President sent a message to Congress recommending the appointment of a competent mixed commission to make a thorough study of the patent law and patent practice and to report its findings to Congress as a basis of legislation.

Your Committee believes that the recommendations of the President, as embodied in his message, are most wise, that there is great danger in any patent legislation that is not based on carefully compiled information, that the appointment of a commission is the only safe procedure, and that the President's recommendations should have the earnest support of all engineers, patentees and others who are interested in the preservation and improvement of our patent system.

Respectfully submitted,

W. H. Blauvelt
B. F. Wood
Committee on Patent Laws

INDEX TO TRANSACTIONS

It is the expectation of the Publication Committee to make this coming season an index of the individual papers in the Transactions, from the first volume up to the present time. It is hoped to make the index a model of its kind, through the coöperation of the librarian of the United Engineering Societies and of others experienced in work of this character.

BOOKS ON AERIAL ENGINEERING

The library of the United Engineering Societies now possesses an important collection of works on aeronautics to which the attention of the membership is called.

At the head of the list may be placed the works of Prof. S. P. Langley (Experiments in Aerodynamics, Washington, 1891, and Memoir on Mechanical Flight, 1911), the dean of the science of aerial engineering. The next place, following the order of development of ideas and not of publication, belongs to the remarkable work of L. P. Mouillard (Le vol sans battement), written sometime in the late eighties or very early nineties of the last century, and clearly establishing the principles of wing warping which gave the mastery of the air to the Wright brothers. A copy of this work was sent to Mr. Octave Chanute in 1894, but was not published until long after the author died, in 1897.

The following works are devoted to the exposition of the fundamental mechanical principles of heavier-than-air flight: Two volumes of Aerial Flight by F. W. Lanchester (Aerodynamics, 1908, and Aerodonetics, 1909), containing a discussion of the theory of aerodynamic support, resistance of bodies in motion to flight, forms of natural flight path, questions of equilibrium, stability in flight, and phenomenon of soaring; Etude de la stabilité de l'aéroplane, by G. de Bothezat (Paris, 1911), examining the necessary and sufficient conditions of the stability of an aeroplane, and for the first time giving a complete mathematical solution of the complicated, but highly important problem of the conditions of stability; Les lois expérimentales de l'aviation, by Alexandre Sée (Paris, 1911), with its companion volume, ordered but not yet received by the library, said to be one of the most remarkable books on the theory of aviation and treating the whole subject in a novel and interesting manner. To the same class belongs the little book on the Force of the Wind, by Prof. Herbert Chatley, dealing with the subject of wind pressure

and stresses in structures due to wind, and giving a brief but clear exposition of the stream line theory, of interest to automobile designers as well as aeronauts. Frederic Houssay, Forme, puissance et stabilité des poissons (Paris, 1912), may also be mentioned here as containing many interesting considerations on the inversion and motion of bodies entirely immersed in a fluid, and although primarily devoted to the investigation of the motion of fish, much of its theory applies also to aeroplanes.

On the margin between pure theory of the art and practical design may be placed Les bases et méthodes d'études aérotechniques, by L. Ventou-Duclaux and Robert, covering both the heavier and lighter-than-air flight, and Victor Lougheed's Vehicles of the Air, the first complete exposition of the whole subject in the English language.

It is the intention of the library to have as complete a collection as possible of the publications of the great European aerodynamic laboratories. So far several publications of the laboratory of Champ-de-Mars, conducted by Mr. G. Eiffel, Honorary Member, Am.Soc. M.E., and of the Laboratorio di Costruzioni Aeronautiche del Battalione Specialisti of the Italian War Office, have been secured, and arrangements have been made to obtain the publications of the Aerodynamic Institute of Riabushinsky at Kuchino, near Moscow, Russia.

On design and construction of heavier-than-air machines the library has the following works: Progress in Flying Machines, by Octave Chanute (New York, 1899), now of historical interest only; The Problem of Flight, by Herbert Chatley (London, 1907), an epitome of the knowledge available on the subject of flight at the time when Santos-Dumont and the Wright brothers were demonstrating that flight is actually possible; Navigating the Air, by the Aero Club of America (New York, 1907); The Art of Aviation, by Robert W. A. Brewer (New York, 1910), containing a large amount of practical information upon the construction of aeroplanes and their motive plant, with a sparing use of mathematics; The Principles of Aeroplane Construction, by Rankin Kennedy (New York, 1911), a concise theory of aeroplane construction and elementary treatment of aeroplane design, with calculations worked out numerically on the two systems in use.

For the lighter-than-air flight the library possesses the following: Espitalier, La Technique du ballon (Paris, 1907); Fr. Walker, Aerial

Navigation (New York, 1902), and A. W. Marshall and Henry Greenly, Flying Machines (London, n. d.).

The library receives the following special periodicals on Aeronautics: Rivista Tecnica d'Aeronautica, Organo dell Aero Club d'Italia; La Technique automobile et aérienne; Zeitschrift für Flugtechnik und Motorluftschiffahrt; Deutsche Luftfahrer Zeitschrift; L'Aérophile; Fly; Aircraft; Aeronautics; and the Aeronautical Journal.

EMPLOYMENT BULLETIN

The Society considers it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is pleased to receive requests both for positions and for men. Notices are not repeated except upon special request. Names and records, however, are kept on current office list three months, and if desired must be renewed at the end of such period. Copy for the Bulletin must be in hand before the 12th of the month. The list of "men available" is made up from members of the Society, with names of other good men not members, and are on file in the Society office. Information will be sent upon application.

POSITIONS AVAILABLE

0194 Note: Published in the September Journal, address should read, Estate of M. Shaik Adam Ipoh, Straits Settlements, or send care of A.S.M.E.

0195 Efficiency engineer for concern in Middle West manufacturing agricultural instruments. Apply through the Society giving age, experience and salary expected.

0196 Shop superintendent, to take charge of finishing department of Pennsylvania concern manufacturing bearing bronze; will be expected to take charge of the power plant and all machinery in the works. Want live, progressive man who can secure the maximum output and work with management in the endeavor to manufacture quickly and cheaply.

0197 Two sales engineers on general line of machinery for New England territory. New York concern.

0198 Superintendent for manufacturing company. Location Michigan. Metal working plant, principally in brass and copper, employing about 1000 men. Prefer experienced and technically educated man; will consider one from any line of manufacturing or railroad work who can show record for efficient handling of labor and economy in operation.

0199 Professor of mechanical engineering for Imperial Pei-Yang University, Tientsin, China; one with several years' experience in teaching and practical work, from 28 to 30 years of age or older. Apply through the Society not later than October 15.

0200 A large manufacturing and engineering company in New York requires a young engineer in the proposition and estimating department, where the opportunities for advancement are exceptionally good. A technical graduate of 1911 or 1912 preferred.

0201 Ohio concern want an engineer, preferably college graduate, with several years' experience in power plant work, particularly in the following up of economies in operation of boilers, and if possible experience with miscellaneous uses of steam in manufacturing processes.

0202 Young engineering graduate for Ohio concern. Position would be with view to acquiring experience and working up to more important work in engineering department.

0203 Engineering salesman with selling record as well as compressed air and hydraulic experience. Location, New York. Apply through the Society.

0204 Assistant to chief engineer in boiler shop department of Pennsylvania concern. Young technical graduate preferred, who after a few months' experience would be competent to get out bills of material for riveted and welded work and able to check drawings for same.

0205 Draftsman on ammonia and brine piping, thoroughly experienced, required by large ice-machine company. State experience, age and salary expected. Apply through the Society.

0206 Wanted: a mechanical engineer with experience in the design of steam and electrical machinery; one somewhat familiar with power plant machinery, who is competent to fill the position of chief engineer in large manufacturing concern. Location eastern Pennsylvania. Apply through the Society.

0207 Mechanical draftsman experienced in general valve construction, or one acquainted with the detail make-up of sanitary plumbing fixtures. A permanent position and good salary to the right party. State salary and experience. Apply through the Society. Location New York.

0208 General superintendent of company in Middle West, manufacturing large engines and special machinery, employing 300 men. Experience required in drafting, machine shop and foundry. Apply through the Society.

0209 Draftsman for position in Morenci, Arizona. Salary $125 a month with opportunity for advancement. One with experience in concentrator work preferred.

MEN AVAILABLE

497 Associate, aged 35, desires position of some responsibility in Philadelphia or suburbs. Sixteen years' experience in civil, structural and mechanical lines, mainly the last, as draftsman, squad foreman, chief draftsman and checker at rolling, steel and pipe mills, by-product coke ovens and on chemical apparatus, steel cars and locomotives.

498 Mechanical and electrical engineer, Cornell graduate, 12 years' experience in design, construction and operation of steam, hydraulic and gas plants for public service and industrial companies, building construction, factory superintendence and appraisals of public service companies, desires permanent position with manufacturing company, consulting engineers or operating company in or near New York City.

499 Member, now general superintendent of machine tool building shop, also experienced in manufacturing electric machinery, engines, and automobiles, desires change. Up-to-date in shop management and methods, machinery and tools. Age 37, married, technical education. Salary $3000.

500 Mechanical engineer, superintendent, chief draftsman or production manager, with practical shop, technical and commercial experience, expert in

the up-to-date manufacture of interchangeable parts, resourceful in design and process for reducing costs and increasing production, good organizer and system man with excellent executive ability, A-1 references, desires permanent position.

501 Junior, technical graduate, aged 28, four years' practical experience in engineering lines, desires a position as superintendent, or assistant superintendent with manufacturing concern.

502 Technical graduate, 24 years of age, at present employed as assistant to superintendent in large water gas plant, desires to make a change to concern having an opening along mechanical or sales engineering lines. Opportunity and scope desired.

503 Engineer, married, 12 years' practical experience in the manufacture of typewriters, adding machines, automobiles, etc., specialist in the design of all kinds of automatic machinery and development of rapid interchangeable manufacturing equipment. Salary not less than $2400.

504 Junior, technical graduate, mechanical engineer, age 30, experience in locomotive shop on combined erecting and repair work and in steel plant; three years' experience in heavy, medium and light machinery including small tool factory, desires position as assistant to a progressive superintendent.

505 Graduate engineer desires position as mechanical superintendent, master mechanic or engineer of construction in power plant work.

506 Mechanical engineer desires position as purchasing agent or business connection with manufacturing firm. Age 28, college training, five years' experience as shop foreman, sales engineer, and contractor. At present with large locomotive works.

507 Technical graduate in mechanical engineering, 25 years of age, 3 years' experience in machine shop and engineering department of large engine manufacturing plant. Desires to become connected with a concern having an opening along mechanical lines.

508 Member, technical graduate, desires position as sales engineer. Experienced as skilled mechanic, chief draftsman and superintendent of construction of power plant equipment. Four years in New York and one in Chicago as sales engineer.

509 Member, with technical education and training and subsequent managerial experience in commercial lines, is open for association and investment, if necessary, with one or more engineers who may have a broad basic patent or device for promotion, development and manufacture.

510 Member, now employed, desires change. Several years' experience as mechanical engineer and consulting engineer for large corporations. Age 54, married, first class references. No objection to location so long as climate conditions will not impair health. Salary $3000 to $4000.

511 Member, at present employed, desires to improve position. Eighteen years' experience especially in internal-combustion engine lines. Qualified for superintendent, assistant manager, designing engineer, or salesman for gas or oil engine concern.

512 Junior, mechanical and electrical engineer, technical graduate, 30 years old, single, experienced in hydro-electric developments, including hydrograph work, design of transmission lines, towers, sub-stations, etc., with operation of same, desires position as chief engineer or designing engineer with large firm of contracting engineers. Best references as to habits and ability. At present employed.

513 Member, hydraulic and electrical engineer, M. I. T. graduate, manufacturing, consulting and operating experience, 42 years old; recently connected with machinery for some of the largest and typical hydraulic projects.

514 Electrical and mechanical engineer open for engagement, Philadelphia or vicinity preferred. Unusual technical training and ability. Experienced in organization of engineering work, research, development of new materials and processes, design of automatic machinery, layout and installation of new factory departments and engineering supervision of manufacturing processes. Exceptional credentials as to character and ability. For further information, address P. O. Box No. 4040, West Philadelphia, Pa.

515 Position as superintendent or other executive position in manufacturing department; 12 years' successful experience in charge of the manufacture of harvesting machines and agricultural implements; 36 years old, married. Best references relative to ability and character.

516 Superintendent or mechanical engineer, Associate member, with 16 years of shop and office experience in design and manufacture of special machinery, soil pipe and fittings, valves, plumbers and steam fitters supplies with proven ability, desires to connect with concern where energy, efficiency, and mechanical knowledge is required of all departments from foundry to shipping. At present employed.

517 Technical graduate, age 30, married, four years as assistant engineer in the manufacture of contractors and railroad equipment, four years engineer on power and industrial plants, railroad car shops and terminals, inspecting, designing, estimating costs and construction, selling. At present employed. Will consider improved position with engineers, contractors, or manufacturers. Personal interview in Philadelphia or New York.

ACCESSIONS TO THE LIBRARY

WITH COMMENTS BY THE LIBRARIAN

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A. I. E. E. and A. I. M. E. can be secured on request from Calvin W. Rice, Secretary, Am. Soc. M. E.

AMERICAN ASSOCIATION OF DEMURRAGE OFFICERS. Proc., 23d Annual Convention. *San Francisco, 1912.* Gift of the association.

DAS AUTOGENE SCHWEISSEN UND SCHNEIDEN MIT SAUERSTOFF, Felix Kagerer. *Wien, 1909.*

BETON UND EISEN. I. Erganzungsheft zum Jahrgang 1912. *Berlin, 1912.*

DIE DAMPFTURBINE ALS SCHIFFSMOTOR, Karl Besig. *Berlin, 1911.*

ELEMENTS OF DRAWING, George F. Blessing and Lewis A. Darling. *New York, J. Wiley & Sons, 1912.*

The work is based on the drawing room courses required of first year students in mechanical engineering in Cornell University. It is intended rather as a reference book than a text book, each drawing plate having certain paragraphs assigned. It includes a somewhat full treatment of isometric drawing.

DIE ESSIG FABRIKATION, Josef Bersch. *Wien, 1907.*

HUMAN FACTOR IN WORKS MANAGEMENT, James Hartness. *New York, McGraw-Hill Book Co., 1912.*

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THE PRESENT STATE OF THE ART OF INDUSTRIAL MANAGEMENT

MAJORITY REPORT OF SUB-COMMITTEE ON ADMINISTRATION

During the past few years a number of striking phenomena, in connection with industrial management, must have become evident even to the most superficial observer. The more important are:

a The widespread, popular interest in the subject which had its rise in a statement made before the Interstate Commerce Commission, in a hearing on the matter of proposed advances in freight rates by carriers. An attorney for the shippers stated on November 21, 1910, that it was estimated that by the application of newly discovered principles of management "in the railroad operation of this country an economy of $1,000,000 a day is possible," and further that these principles can be applied with equal success "in every form of business activity." This popular interest is shown by the great number of articles published in the daily papers and popular magazines, mediums that give but scant attention to technical subjects, except of the most striking nature.

b The suddenly intensified interest in the subject on the part of employers and business executives in many lines of activity, shown by lectures, addresses, professional papers and reports presented to their associations.

c The opposition of labor unions to the newer methods of management, shown by statements of labor leaders, in a few instances by strikes, and by an attempt to

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prohibit by law the use of some of these methods in Government shops.

d Governmental recognition of the matter shown by the appointment of a special committee of the House of Representatives to investigate systems of management in Government arsenals and shops, which reported in March 1912; by the appointment of a civilian board by the Secretary of the Navy to investigate management in the navy yards, which reported in July 1911; and by Senate bill *S 6172*, now in committee, which is intended to prohibit time study and the payment of premiums or bonus on Government work.

e The rapidity with which literature on the subject has accumulated. One directory of books on business management lists 500 titles, and states that 75 per cent of them have been written within five years.

f The formation of two societies having as an aim the furtherance of the application of the principles of management.

g The separation of persons interested in the matter into two camps, one of enthusiastic advocates, the other of vigorous opponents of what is called the new element in management.

h The unquestionable proof of the advance that can be made in unskilled work, as shoveling material, and in ancient trades, as bricklaying, by the application of the principles of management. This is the most striking phenomenon of all.

THE PRINCIPLES OF MANUFACTURE

2 Before defining the element in the art of management that has given rise to these phenomena, it is necessary to review briefly the beginnings of modern industry. This gives a historical setting from which the present can be more truly judged.

3 Modern industry is stated by some writers to have begun in 1738 when John Wyatt brought out a spinning machine. Others place the period as between 1750 and 1800, when the power loom and steam engine came into being. It was marked by the development of labor-saving machinery. It was brought about by the change from handicraft to manufacture.

4 Early British economists held that the application of the principle of *division of labor* was the basis of manufacture. From Adam Smith's "Wealth of Nations," 1776, we quote:

"This great increase of the quantity of work which, in consequence of the division of labor, the same number of people are capable of performing, is owing to three different circumstances; first, to the increase of dexterity in every particular workman; secondly, to the saving of the time which is commonly lost in passing from one species of work to another; and lastly, to the invention of a great number of machines which facilitate and abridge labor, and enable one man to do the work of many."

5 Charles Babbage, the great British mathematician and mechanician, believed that from the above-quoted statement the most important principle was omitted. This omission he supplied as follows in his "Economy of Machinery and Manufacture," 1832:

"That the master manufacturer, by dividing the work to be executed into different processes, each requiring different degrees of skill and force, can purchase exactly that precise quantity of both which is necessary for each process; whereas, if the whole work were executed by one workman, that person must possess sufficient skill to perform the most difficult, and sufficient strength to execute the most laborious, of the operations into which the art is divided."

6 It appears, however, that another principle is the basic one in the rise of industry. It is the *transference of skill*. The transference of skill from the inventor or designer to the power-driven mechanism brought about the industrial revolution from handicraft to manufacture. It will be necessary to refer to this principle frequently throughout this report, in showing the meaning and position of management in industry.

7 No better single illustration of the application of this principle can be found than in the invention of the lathe slide rest by Henry Maudsley in 1794. This has been ranked as second only to the steam engine in its influence on machinery building, and thus on industrial development. The simple, easily controlled mechanical movements of the slide rest were substituted for the skilful human control of hand tools. So complete has been this transference of skill that today hand tooling is a vanished art in American machine shops. Very few lathe hands can chase a thread with hand tools, yet all can cut good threads on an engine lathe, thanks to the slide rest. After the traditional skill of a trade, or the special, pe-

culiar skill of a designer or inventor, has been transferred to a machine, an operator with little or no previously acquired skill can learn to handle it and turn off the product.

8 An example of the extent to which this transference of skill is carried today is presented by the shoemaking industry. The United Shoe Machinery Company builds some 400 machines used in shoe manufacture. These are so highly organized that the greater part of shoe-shop operatives are unskilled except in a single readily mastered detail of the work. The skill in shoemaking is now in the mechanical equipment of the shops. This transference is a development of the past 50 years.

9 James Nasmyth, a British engineer, inventor of the steam hammer, has this to say in 1851 of the application of this principle in his own works: "The characteristic feature of our modern mechanical improvements, is the introduction of self-acting tool machinery. What every mechanical workman has now to do, and what every boy can do, is not to work himself, but to superintend the beautiful labor of the machine. The whole class of workmen that depend exclusively on their skill is now done away with."

10 Methods of analyzing and recording operations were early developed. Adam Smith records the divisions of the work of manufacturing pins, listing 11 operations. Charles Babbage gives a table (see Appendix No. 1, Table 1) from a French investigator, showing the number of operations, time for each, cost of each, and expense of tools and material for making pins in France in 1760. He gives a similar table for English manufacture in his day (see Appendix No. 1, Table 2, from Economy of Machinery and Manufacture, 1832).

11 He further comments on the use of the watch to time operations. We quote from his instructions to one making such observations and using a skeleton form that he recommends: "In filling up the answers which require numbers, some care should be taken: for instance, if the observer stands with his watch in his hand before a person heading a pin, the workman will almost certainly increase his speed, and the estimate will be too large.....The number of operations performed in a given time may be frequently ascertained when the workman is quite unconscious that any person is observing him. Thus the sound made by the motion of a loom may enable the observer

to count the number of strokes per minute, even though he is outside the building in which it is contained."

12 M. Coulomb, the noted French physicist (1736-1806), who had great experience in making such observations, cautions those who may repeat his experiments against being deceived by such circumstances. We translate a single quotation: "I pray (says he) those who wish to repeat them (the experiments) if they have not time to measure the results after several days of work, to observe the workmen at various times during the day without their knowing that they are being watched. We cannot be too well warned of the danger of self-deception in computing either the speed or the effective time of work through an observation of a few minutes."

13 Thus we see the application of the principle of *transference of skill* at the basis of the development of the industry, and an early appreciation of the value of the detailed study of operations in making that transference more complete. But the machine was the viewpoint. It was looked upon as the producing unit. Combined and contrasted with this was a lack of knowledge of scientific principles and their sure application. Charles Babbage treats of this forcefully. We quote:

> "There is perhaps no trade or profession existing in which there is so much quackery, so much ignorance of the scientific principles, and of the history of their own art, with respect to its resources and extent, as is to be met with amongst mechanical projectors."

14 In the same vein he emphasizes the need of accurate drawings as if having in mind the poor quality of the work from the average draftsman of his day: "It can never be too strongly impressed upon the minds of those who are devising new machines (says he) that to make the most perfect drawings of every part tends essentially both to success of the trial, and to economy in arriving at the result."

15 He further points out that there is another important factor in successful industry, in addition to machinery. We read that "in order to succeed in a manufacture, it is necessary not merely to possess good machinery, but that the domestic economy of the factory should be most carefully regulated."

16 These quotations foreshadow modern methods of thinking out the work in advance and transferring this thought to the workmen. The subsequent development has had the effect

of advancing still further the division of labor, and beginning the division of thought. The drafting room presents the first example of the trend, in its collection of engineering data, in its prediction of results and the formation of staff organization.

17 But from the period of the last quotation almost to the present there has been no change in the basic principles discovered and applied in industry. There has been nothing but an extension of those already known. The place of greatest advance has been in the drawing room. The art of machine design has been greatly developed. The last half of the last century saw a tremendous increase in inventions, a tremendous furtherance of the application of transference of skill to machines and tools. The skeleton of an industrial organization of this period, one that was too large for a single executive to manage, consisted of a designing department and a production department, each with a head responsible to the manager.

18 The first of these, the one that was the means of embodying skill in the machinery and tools of production, was highly developed and organized. Experiment, research and detailed study were constantly resorted to, to aid in reaching the desired result. The work was highly specialized and the employees highly paid. Not infrequently the manager or chief executive devoted much of his own time to this part of the business.

19 The production department presented a contrasting condition. The workmen were given the tools and machines designed in the drawing room and using their own unaided skill were expected to produce work of the desired quality and quantity. Except in rare instances no effort was made to transfer the skill of the management to the production department and the employees, or to undertake the division of executive thought. Very little consideration was given to the workmen as a producing unit.

FEATURES OF THE CHANGE

20 Within the past 20 or 25 years certain changes have taken place in the attitude of many production managers toward the problems that they face and the forces and means that they control. An increasing amount of attention is being given to the worker. An early evidence was the development of profit-sharing, premium and bonus systems to reward increased effort

and output. There followed welfare work, industrial betterment movements, the adoption of safeguards and regulations to minimize industrial accidents, the substitution of the principle of accident compensation for employers' liability and an improvement in the physical surroundings and conditions of factories. All of these tendencies have been fostered and to a great extent initiated by employers. But even today these are by no means generally adopted.

21 Another tendency, less pronounced in character, has as its object the improvement of the personal relations between employé and employé and between employé and employer. It is an effort to establish the best of factory working conditions in those things not physical in nature, to develop and maintain a shop atmosphere free from all harassing and hindering influences. It is an attempt to make use of the results of experimental psychology, in improving working conditions.

22 But the most important change and one that comprehends the others, is in the mental attitude toward the problems of production. The tendency is toward an attitude of questioning, of research, of careful investigation of everything affecting the problems in hand, of seeking for exact knowledge and then shaping action on the discovered facts. It has developed the use of time study and motion study as instruments for investigation, the planning department as an agency to put into practice the conclusions drawn from the results of research, and methods of wage payment which stimulate coöperation.

23 All of these changes have affected the production department much more than the designing department. The effect is to extend the principle of transference of skill to production, so that it completely embraces every activity in manufacture. The skill of the management is consciously transferred to all of the operations of the factory. This extension is expressed by these phrases: the drawing room is the planning department of design, and the planning department is the drawing room of production.

NATURE OF THE COMMITTEE'S INVESTIGATION

24 To obtain information on present conditions your committee wrote to the recognized experts, to executives of plants in many lines of industry, to students of industrial problems,

and has had many interviews with men in these various fields. The response to our requests has been in the main most generous. We are deeply indebted to the information thus received for a large portion of the following sections of this report. We are glad to take advantage of this opportunity to express our gratitude to all those who have given aid.

25 Throughout the following pages there is a plentiful use of illustrative quotations. Many of these are taken from correspondence resulting from our investigations. Others are from the mass of literature mentioned in Paragraph 1*e*.

26 On some points diametrically opposed views have been expressed. In such cases we have presented both. In no case has credit been given for these views or quotations, as the information was solicited in confidence.

DEFINITION OF THE NEW ELEMENT IN THE ART OF MANAGEMENT

27 Requests for a definition of the new element in the art of management brought forth a difference of opinion as to its existence. The opposed view is given in the following quotations:

"I am not aware that a *new* element in the art of management has been discovered———."

"There have been no new discoveries in scientific management of industrial institutions. Common-sense men have used common-sense methods always. The term 'scientific management' is a catch-word which assumes that industrial institutions have not been scientifically managed—which is not the case. My experience and the experience of my friends has been that there has been no new element injected into the art of management."

"In the writer's opinion there is very little that is new about it (the art of management). There is hardly any part of it that has not been practised by managers for the past 100 years. The trouble is there are not enough managers with sufficient initiative to set the system moving properly."

"——— the problem presented is not the adoption of something entirely new; but rather the extension to every detail of our work of something which we have already tried."

28 Turning now to the other side of the question, from a large number of definitions of this new element we select the following as very nearly conveying, taken together, the complete conception as our investigation has disclosed it:

"The best designation of the new element I believe to be 'scientific management.' This term already has been adopted quite generally and although frequently misused, carries with it the fundamental idea that the manage-

ment of labor is a process requiring thorough analytical treatment and involving scientific as opposed to 'rule of thumb' methods."

"The writer ventures to define the new element briefly, but broadly, as: The critical observation, accurate description, analysis and classification of all industrial and business phenomena of a recurring nature, including all forms of coöperative human effort and the systematic application of the resulting records to secure the most economical and efficient production and regulation of future phenomena."

"Stripped of technicalities the method of the modern efficiency engineer is simply this: First, to analyze and study each piece of work before it is performed; second, to decide how it can be done with a minimum of wasted motion and energy; third, to instruct the workman so that he may do the work in the manner selected as most efficient."

"The Taylor System is not a method of pay, a specific ruling of account books, not the use of high-speed steel. It is simply an honest, intelligent effort to arrive at the absolute control in every department, to let tabulated and unimpeachable fact take the place of individual opinion; to develop 'team play' to its highest possibility."

"As we conceive it, scientific management consists in the conscious application of the laws inherent in the practise of successful managers and the laws of science in general. It has been called management engineering, which seems more fully to cover its general scope than a science."

29 These quotations convey the ideas of a conscious effort to ascertain and study facts and systematically to apply them in instructing the workmen and in controlling every department of industry. Setting these against the underlying principle of the transference of skill we conceive the prominent element in present-day industrial management to be: *The mental attitude that consciously applies the transference of skill to all the activities of industry.*

30 Here emphasis is placed upon the word *all* for, as shown in Pars. 17 and 18, the restricted application of this principle to machines and tools has been highly developed for a long period. But its conscious application in a broad way to the production departments, and particularly to the workmen, we believe has been made during the last quarter century.

RISE OF THIS MENTAL ATTITUDE

31 The rise of this change of attitude in regard to industrial management is shown in the papers on the subject in the Transactions of this Society. These are 16 in number and are listed in Appendix No. 2 of this report. The period covered is from 1886 to 1908. The practice upon which several were based

extended over a number of years before the paper was presented. Papers on accounting have been excluded.

32 The first, No. 207, classifies *management of works* as a modern art having a vast amount of accumulated experience, points out that the executives must have "a practical knowledge of how to observe, record, analyze and compare essential facts in relation to ..... all..... that enters into or affects the economy of production and the cost of the product," and makes a plea for the interchange of management data.

33 Eight following papers, Nos. 256, 341, 449, 596, 647, 928, 965 and 1012, deal with methods of wage payment, showing the increasing attention given to the workmen during this period. Of these methods the "premium plan" described in paper No. 449 has an extensive use today in machine shops. It probably ranks third, in extent of use, being exceeded by day work and piece work in the order named. Paper No. 647 outlines elementary rate fixing; that is, the minute study of each detail of each operation. From this, motion study and time study have grown. The "bonus system" of paper No. 928 also has an extensive use, probably ranking fourth.

34 Paper No. 1003, "Shop Management," is the first complete presentation of the subject. This paper with the subsequent writings of its author, stands today as the only comprehensive outline of industrial management. Papers Nos. 1001, 1002, 1010, 1011 and 1115, are amplifications of certain features of No. 1003 and are based on the same practice.

35 Paper No. 1221 deals with the training of workmen, and outlines practical, tested methods of bringing about the all-important transference of skill.

LABOR-SAVING MANAGEMENT

36 Since these papers were presented, and during the development of popular interest in the subject, the term "scientific management" has been generally and loosely applied to the new system and methods. This is commonly taken to mean that there is a science rather than an art of management. A truer interpretation is that it means management using scientific methods, these being taken largely from the sciences of physics and psychology.

37 The expression "labor-saving management" better con-

veys the meaning of the movement. It has the further advantage of being easily and surely understood because of its strict analogy with the term "labor-saving machinery." It is no chance that puts these two terms labor-saving machinery and labor-saving management, in conjunction, for the first is the past development and the second the present trend of industry, and they will be closely and inevitably associated in the successful manufacturing of the future. Throughout the following pages of this report the terms "industrial management" and "labor-saving management" are used, the first to denote the subject broadly, the second the newer attitude.

THE REGULATIVE PRINCIPLES OF INDUSTRIAL MANAGEMENT

38 The lack of accurate thinking and clear expression in regard to management are nowhere better shown than in many of the statements of the so-called principles. These can be divided into two classes, personal characteristics of managers and mechanical means of applying. It is evident that neither can show us the way in which the activities of industry are to be regulated.

39 In our investigation preparing for this report, one correspondent writes as follows:

"The regulative principles of management along scientific lines include four important elements:

a Planning of the processes and operations in detail by a special department organized for this purpose.

b Functional organization by which each man superintending the workman is responsible for a single line of effort. This is distinctly opposed to the older type of military organization, where every man in the management is given a combination of executive, legislative and judicial functions.

c Training the worker so as to require him to do each job in what has been found to be the best method of operation.

d Equable payment of the workers based on quantity and quality of output of each individual. This involves scientific analysis of each operation to determine the proper time that should be required for its accomplishment and also high payment for the worker who obtains the object sought."

40 Another correspondent finds the solution of problems of management in the observing and regulating of three classes of industrial phenomena:

"*a* The economic results of different arrangements and forms of

materials and operations upon them, either to produce equipment or product. This covers the whole field of recorded experience from invention and design of product and tools down through the successive shop processes to ultimate finished product and its tests in service. It is the object of the scientific method to make the best of this experience, in its essential details, readily available for all concerned, and to see that it is actually absorbed and put in practice.

b The economic results of varying executive methods for effectively directing human efforts as a whole in the use of the above experience. This covers the entire field of building up, coördinating and controlling the supervising organization of a plant with its statistical and recording systems.

c The economic results of steps taken to raise the industrial efficiency of the individual worker in every grade of service. This covers the whole problem of labor reward, intensified ability, conserved energy and the general relations of employer and employée."

41 We have pointed out that the underlying principle, that is, cause in the widest sense, the application of which has built up modern industry, is the transference of skill. This basic principle is put into effect on the management side of all industrial activities, through three regulative principles which sum up the ideas in the above quotations, Pars. 39 and 40. These have been concisely stated as[1]: (*a*) the systematic use of experience; (*b*) the economic control of effort; (*c*) The promotion of personal effectiveness.

42 The first includes the use, in all essential detail, of traditional knowledge, personal experience and the results of scientific study on the part of the executive force. It implies the accumulation and use of records and the setting up of standards.

43 The second includes the division and subsequent coördination of both executive and productive labor; the planning of single lines of effort, the setting of definite tasks and the comparison of results; and the effective training of the workers. It implies the previous acquisition of skill by the executives.

44 The third includes a definite allotment of responsibility and the adequate, stimulative encouragement and reward of both executive and productive labor; the development of contented workers, and the promotion of their physical and mental health. It implies the most thorough comprehension of the human being.

[1] American Machinist, vol. 36, p. 857, The Principles of Management, by Church and Alford.

THE PRACTICE OF MANAGEMENT

45 As labor-saving management springs from a change in mental attitude, the beginning of its practice should be with the persons having the final responsibility, the proprietors of closely-owned businesses, the directors of larger establishments, or the officials having charge of Government works. Before any changes are made, such men should clearly understand the viewpoint from which all of the managerial work is to be done, the principles that are to be applied, the general method of their application and the results expected.

46 A similar mental attitude must be fostered among all the members of the executive force and a period of training for them begun. This may include a redistribution of function and responsibility, and will include a detailed study of production by scientific methods. This is the period of division of thought, training of the management staff and setting-up standards of performance. This must be carefully performed before there can be effective transference of skill to the workers in the production departments.

47 The usual conception of modern management is that it affects the workmen most of all, tending to stimulate them to turn out increased production to their possible hurt. This is wrong. If the principles outlined are followed, the executive, or non-producing labor is the most affected. Its individuals are compelled to study, plan and direct. They must acquire knowledge and skill in order to transfer it. It is a system of management that forces the executives to manage.

48 This being so, the introduction of modern management in a plant must be made slowly. The causes of most so-called failures are principally two: a failure of the executives to acquire the vital mental attitude and too great haste in application. The latter seems to be the dominant one. Your committee feels compelled to emphasize the danger of attempting to hurry any change in methods of management. Each step of the work should be made permanent before the next is begun.

49 We have examined records of production which clearly show a lessening of individual output among workers who had been trained for some time and had achieved good results as soon as untrained workers were put with them, thus lessening their share of personal supervision. Later the original standard

of production was again reached, but the results seemed to be directly proportional to the amount of skilful supervision, during a lengthy period of training.

50 After those who are to operate the new methods have acquired the necessary knowledge and established sufficient standards, the work of putting these into effect can be begun. This means the fixing of the best attainable working conditions and giving each worker definite tasks with an adequate reward to each one who attains to the standard set. This part of installing the methods must be accomplished with tact and patience, remembering that leadership and example are powerful aids in bringing about enthusiastic coöperation.

51 The training of the workers is essential in this part of the application. This must be far more than mere demonstration, the mere showing that a thing can be done. It must be patient teaching and help until the required degree of dexterity or skill is acquired, that is, up to the habit stage. It is evident that such work cannot be hurried.

52 Such, broadly, are the three steps in the practice of management. It is now necessary to investigate the internal elements of permanence in such methods. If the proper mental attitude is once taken, we believe it will never be given up. This is substantiated by a few cases when early attempts to improve management were failures and the methods abandoned. Later, however, other attempts were made with substantial success. The mental attitude outlived the failure. Thus in a given industrial organization this feature would not be lost except by a loss of the executive staff.

53 The permanence of records of performance and standards needs only to be mentioned to be appreciated. Once set up in an industry, disaster is invited if they are disregarded.

54 To these is added a third in the nature of a spur from the working forces to the managing force. An adequate reward is one of the essentials. Whatever disturbs the mechanism of production interferes with the earning of the rewards. The workers at once object, pointing out the trouble and insisting that it be rectified. The management is spurred to keep all conditions up to the fixed standard. Examples of this action have been brought to the attention of your committee.

55 The practice as outlined, while built upon fixed standards

and procedure, is by no means rigid and inflexible as has been alleged. The design and construction of labor-saving machinery is carried on with a multiplicity of different details. Labor-saving management should likewise use a variety of details suited to the requirements of different industries and plants. There can be nothing fixed in such human endeavor except the underlying principle. As a simple matter of fact we have found different methods, details and nomenclature in use in different plants. Many efforts have undergone marked change and development since first installed. Further, this idea of rigidity is repudiated by some of the foremost management experts.

56 In Par. 39 is emphasized the need of a scientific study of everything connected with production. The methods used are adapted from the research laboratory. But the purpose of their use is changed. The scientific investigator uses his laboratory to discover facts. Their discovery and declaration is his end and aim. The management investigator uses laboratory methods to discover facts for immediate use. The end and aim is utility. This is the test of industry. It is therefore unwise and in fact detrimental to carry investigations to an extreme. Enough facts must be observed to shape intelligent action. Persons having time study and motion study in charge should possess that rare, intuitive, human quality that causes its possessor to know when enough observations have been collected to form a sound working conclusion.

57 The position of the expert in the practice of management is more clearly seen as experience increases. The element of mystery has already departed. This is to be welcomed for it means the downfall of mere "systematizers." One of the unfortunate features of this great movement has been the rise of alleged experts who have been ready to promise extravagant results if they were allowed to systematize an industrial plant. The test which their work cannot meet is the one of permanence.

58 An industrial manager who has had signal success in directing large enterprises sums up the more undesirable characteristics of systematizing practice as:

> "*a* The publication and quotation of statistics regarding gains made through the use of particular systems, without a frank statement of the degree of inefficiency of the plants before reorganization.

b The failure to view the plant from the investor's standpoint rather than as a laboratory offering opportunities for interesting and expensive experience.

c The failure to admit that every application of past solutions to unstudied new and different conditions *is* an experiment.

d The waste of time and money on problems that will yield to scientific treatment, but which do not recur often enough to justify such a solution.

e The undervaluing of effective leadership in management and consequent lack of permanency in results.

f The overvalue of emasculated 'system' leading to a curious nonresponsibility on the part of any person for the total result.

g The frequent assumption that the treatment of the problems of similar plants should be identical.

h The failure to properly appraise in a growing concern the value of its internal asset of 'good will.'

i The imperfect analysis and appreciation of the human factor in industry, with a consequent failure to reckon patiently with 'habit' and 'inertia' and a tendency to hasty 'substitution,' bringing about the breaking up of valuable organization."

59 The real expert concentrates on the facts of a given problem, and from a wide experience in analysis, coördination and practical responsibility works out a solution by scientific methods, suited to the material and human factors involved. The tendency is for him to do less of the detail work of installation, but to train and direct the persons who are permanently to manage. This is a true process of transference of skill.

STATISTICAL DATA

60 Your committee hoped to present statistics on the extent to which labor-saving management is in use. This could not be realized. Many industrial managers whom we have addressed have not honored us with their confidence in this direction. In fact, it seems as if a secretive stage is now with us. There are two reasons for withholding such information. The first is identical with the one that has developed "trade secrets" and secretiveness in regard to machines, tools and processes, the desire to keep things of value away from competitors. The second is a belief that in the minds of some persons a reflection is cast upon the ability of the executives of an industrial establishment if outside experts are employed. Frequently a system of management is referred to as the development of some one in the organization, although it was installed by a management expert, employed for the purpose.

61 Some idea of the variety of the industries in which labor-saving management is in use can be gained from Appendix No. 3, which lists a total of 52.

BROAD RESULTS OF LABOR-SAVING MANAGEMENT.

62 In cases where the use of labor-saving management can be considered a success, the broad results have been: a reduced cost of product; greater promptness in delivery with the ability to set and meet dates of shipment; a greater output per worker per day with increased wages; and an improvement in the contentment of the workers. This last item is shown by the fewness of strikes under the new management, and in the refusal of those working under the changed conditions to join in a strike of their fellows in the same plant who were not working under the new methods. This last-mentioned situation has arisen a number of times. In one case an attempt was made to strike a room where about one-half of the operators were under the new conditions. These refused to go out; the rest went.

63 These results indicate certain advantages to both employer and employé. But it is charged that the movement has not yet entirely justified itself from the economic viewpoint, for it has not reduced the cost of product to the consumer. The implication is that its possibilities will not be realized until employers, employés and the public are alike benefited. With this view we are in most hearty accord. Labor-saving machinery has brought the comforts that we all enjoy today. Labor-saving management promises to extend those comforts. Where properly administered it is conserving labor and is thus contributing to the good of society at large, and although the benefit to the consumer may not yet be generally felt, it has already developed to a certain extent and will continue to develop as the natural result of increased production.

J. M. DODGE, *Chairman*
L. P. ALFORD, *Secretary*
D. M. BATES
H. A. EVANS
WILFRED LEWIS
W. L. LYALL
W. B. TARDY
H. R. TOWNE

Members Sub-Committee on Administration

APPENDICES

APPENDIX NO. 1

64 The following tables are taken from a book published in 1832, Economy of Machinery and Manufacture, by Charles Babbage. Table 1 gives the cost and operations in detail in manufacturing 12,000 No. 6 pins in France in 1760. The observations were made by M. Perronet.

65 Table 2 gives similar data for manufacturing pins in England in the time of the author. The size is "eleven" of which there are 5546 to the lb.

TABLE 1 OPERATIONS IN PIN MANUFACTURE IN FRANCE IN 1760

	Name of the Process	Time of Making 12,000 Pins, Hr.	Cost of Making 12,000 Pins, Pence	Workman usually Earns per Day, Pence	Expense of Tools and Materials, Pence
1	Wire	....	....	....	24.75
2	Straightening and Cutting	1.2	0.5	4.5	....
3	Coarse Pointing	1.2	0.625	10.0	....
	Turning Wheel *	1.2	0.875	7.0	....
	Fine Pointing	0.8	0.5	9.375	....
	Turning Wheel	1.2	0.5	4.75	....
	Cutting off Pointed Ends	0.6	0.375	7.5	....
4	Turning Spiral	0.5	0.125	3.0	....
	Cutting off Heads	0.8	0.375	5.625	....
	Fuel to Anneal Heads	....	....	....	0.125
5	Heading	12.0	0.333	4.25	....
6	Tartar for Cleaning	....	....	....	0.5
	Tartar for Whitening	....	....	....	0.5
7	Papering	4.8	0.5	2.0	....
	Paper	....	....	....	1.0
	Wear of Tools	....	....	....	2.0
		24.3	4.708		

* The expense of turning the wheel appears to have arisen from the person so occupied being unemployed during half his time, whilst the pointer went to another manufactory.

TABLE 2 OPERATIONS IN PIN MANUFACTURE IN ENGLAND A BOUT 1830

	Name of the Process	Workmen *	Time of Making 1 Lb. of Pins, Hr.	Cost of Making 1 Lb. of Pins, Pence	Workman Earns per Day	Price of Making Each Part of a Single Pin in Millionths of a Penny
					s. d.	
1	Drawing Wire	Man	0.3636	1.2500	3 3	225
2	Straightening the Wire	Girl	0.3000	0.1420	0 6	26
		Woman	0.3000	0.2840	1 0	51
3	Pointing	Man	0.3000	1.7750	5 3	319
4	Twisting and Cutting the Heads	Boy	0.0400	0.0147	0 4½	3
		Man	0.0400	0.2103	5 4½	38
5	Heading	Woman	4.0000	5.0000	1 3	901
6	Tinning, or Whitening	Man	0.1071	0.6666	6 0	121
		Woman	0.1071	0.3333	3 0	60
7	Papering	Woman	2.1314	3.1973	1 6	576
			7.6892	12.8732		2320

* Number of Persons Employed: Men, 4; Women, 4; Children, 2. Total, 10.

APPENDIX NO. 2

66 Following is a complete list of papers published in the Transactions of The American Society of Mechanical Engineers dealing with industrial management:

No.			
207	The Engineer as an Economist	Henry R. Towne	1886
256	A Problem in Profit Sharing	Wm. Kent	1887
341	Gain Sharing	Henry R. Towne	1889
449	The Premium Plan of Paying for Labor	F. A. Halsey	1891
596	The Relation of the Drawing Office to the Shop in Manufacturing	A. W. Robinson	1894
647	A Piece-Rate System	Fred. W. Taylor	1895
928	A Bonus System for Rewarding Labor	H. L. Gantt	1902
965	Gift Propositions for Paying Workmen	Frank Richards	1903
1001	The Machine-Shop Problem	Charles Day	1903
1002	A Graphical Daily Balance in Manufacture	H. L. Gantt	1903
1003	Shop Management	Fred. W. Taylor	1903
1010	Slide Rules for the Machine Shop as a Part of the Taylor System of Management	Carl G. Barth	1904
1011	Modifying Systems of Management	H. L. Gantt	1904
1012	Is Anything the Matter with Piece Work?	Frank Richards	1904
1115	A History of the Introduction of a System of Shop Management	James M. Dodge	1906
1221	Training Workmen in Habits of Industry and Coöperation	H. L. Gantt	1908

APPENDIX NO. 3

67 Following is a list of the industries in which some form of labor-saving management has been installed:

Book binding
Building construction
Carriage and wagon building
Construction and repair of vessels (navy yards)
Fire-arms and ordnance
  Rifles
  Gun carriages
Machinery building
  Automobiles
  Agricultural implements
  Coal-handling machinery
  Electrical machinery
  Founding, iron and brass
  General machine work
  Gas engines
  Locomotives
  Machine tools
  Molding machines
  Pumps
  Pneumatic toools
  Sewing machines
  Typewriters
  Wood-working machinery
Metal and coal mining
Metal working
  Bolts and nuts
  Chains
  Hardware
  Tanks
  Tin cans
  Valves and pipe fittings
Miscellaneous manufacturing
  Beer
  Beet sugar
  Boxes (wood and paper)
  Buttons
  Clothing
  Cordage
  Food products
  Furniture
  Flour
  Glass
  Lumber products
  Pianos
  Paper and paper pulp
  Rubber goods
  Soaps
  Shoes
  Slate products
Printing and lithographing
Railroad maintenance of motive power
Steel manufacture
Textile manufacture
  Bleaching and dyeing
  Cottons
  Velvets
  Woolens

THE PRESENT STATE OF THE ART OF INDUSTRIAL MANAGEMENT

MINORITY REPORT OF SUB-COMMITTEE ON ADMINISTRATION

I am unable to sign the majority report in its entirety, much as I admire the thoroughness with which it has been prepared and its great interest. In its general tenor it distinctly implies the desirability of what is termed labor-saving management, involving the planning department, functional organization and the bonus system. Perhaps this statement is not strictly justified, but I cannot avoid the impression after reading the report most carefully. That the methods of management have undergone a great change in recent years I certainly agree. I would explain it by stating that in many respects the art of management is developing into a science. In common with most lines of work, the method of investigating facts has changed. Phenomena are analyzed, information is obtained accurately, the mental attitude of the manager is scientific rather than empirical. Things that used to be known generally by gradual experience are now known specifically by detailed observation. In the course of this development many new systems and ideas have been invented, time studies, motion study, payment systems, functional management, etc. Some are new, others partly new, others simply practices of many managers put into definite form. The science of management will classify these for us, perhaps even explain their advantages and limitations so that we may in time know which is preferable and when.

2 But the introduction of the use of these methods has been attended by claims as to the results that might be obtained by their use, and these claims have led to such absurd statements as that made before the Interstate Commerce Commission, which is referred to in the report. I feel most strongly that each of the suggestions that have been made to improve methods of management may have merit, but I do not feel that any one of them is a panacea for all our inefficiency. For instance, in certain classes of work time studies are valuable, in others they may be a waste of time. In certain

classes of work I consider piece-work or bonus systems desirable, in others I consider them inferior to day work. Functional management may be an improvement in certain industries, in others I do not consider it suitable. In some cases the workman can be trained by the skilled executive, in others he can train him with ease.

3 In general I feel that labor-saving management is not any particular system, but will always remain the art of selecting and applying the most appropriate methods furnished by the science of management, the science that records what these methods are and the results obtained from them. As you will see, I agree with your correspondent in his last paragraph, quoted under Par. 28; but his opinion is not really represented in the trend of the report.

H. H. VAUGHAN
Member, Sub-Committee on Administration

AIR IN SURFACE CONDENSATION

By Geo. A. Orrok

ABSTRACT OF PAPER

This paper is an addition to the author's paper on the Transmission of Heat in Surface Condensation and gives the results of his investigations on the amount of air present in feedwater and the presence and effect of air in surface condensers, and describes an apparatus for measuring the air leakage.

AIR IN SURFACE CONDENSATION

By Geo. A Orrok, New York

Member of the Society

In the paper on The Transmission of Heat in Surface Condensation[1], the subject of the effect of air on vacuum and heat transference was considered and reference was made to the few authorities who had investigated this subject. The author was unable to find that any one had quantitatively measured the air in a condenser, except J. A. Smith who had introduced known quantities of air and noted the effect on vacuum and heat transference.

2 In view of the lack of knowledge of this subject the author determined to investigate:

a The amounts of air contained in feedwater and those entering the condenser through the prime mover.

b The amounts of air removed from the condenser by means of the air pump.

c The effect of known quantities of air on the action of a surface condenser of commercial size.

3 The problem of air in the feedwater has been investigated by Smith whose paper appeared in London Engineering, October 7, 1904. He gives a table showing that the gases present in various samples of fresh and salt water varied from approximately 7/10 of one per cent to as much as 3½ per cent by volume. He also investigated the emission of gas from water when heated to about the boiling point and gave figures and curves showing his results.

4 The author's investigations were made at the Waterside No. 2 station of the New York Edison Company and the water samples were taken both from the suction and discharge sides

[1] Trans. Am.Soc.M.E., vol. 32, p. 1139.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

of the feed pump. The average temperature of the feedwater during the tests, after having passed the heater, was about 190 deg. fahr. It was thought possible that there might be air bubbles mechanically entrained in the water, besides the amount of air in solution. To determine the amount of this entrained air samples of the water were allowed to flow through the apparatus which is shown in Fig. 1. The water entered through

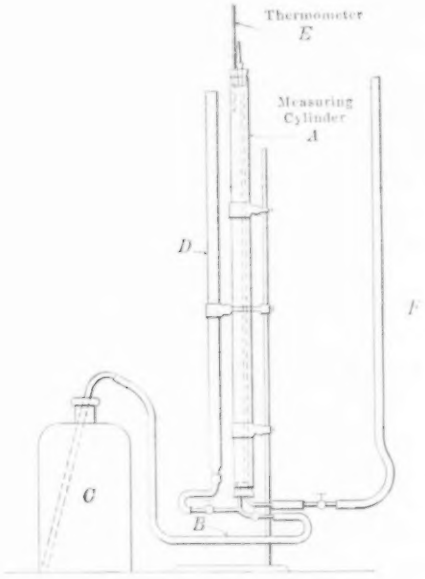

FIG. 1 APPARATUS USED TO DETERMINE AMOUNT OF AIR MECHANICALLY ENTRAINED IN WATER

the tube *F* and passed to the top of the measuring cylinder where the air separated and was collected at the top of the cylinder; the water flowed out of the cylinder through the tube *B* into the glass demijohn where it was measured. These tests were started with the apparatus entirely filled with water. The air collected was measured at atmospheric pressure by means of the pressure-regulating cylinder *D*. The temperature of the air was determined by the thermometer *E*. The amount of air dissolved in the feedwater was determined chemically by means of the following reactions:

$$2Mn(OH)_2+O_2=2H_2MnO_3$$
$$2H_2MnO_3+8Hcl=2MnCl_2+6H_2O+2Cl_2$$
$$4KI+2Cl_2=4KCl+2I_2$$

5 The free iodine liberated was titrated with a solution of sodium thiosulphate of known strength and its amount determined. From this amount the quantity of oxygen in solution in the sample of water taken was calculated. The amount of air in solution was then figured from the known ratios of oxygen and nitrogen in solution in water at various temperatures. The determination in this way may be considered quite accurate.

6 This determination of air in water was also made of a number of samples of Croton water (the make-up feed) and from hot well water. The following results, in per cent, were obtained of air by volume at atmospheric pressure:

	187 deg. fahr.
Mechanically entrained in feedwater, average of ten tests	0.0151
In solution in feedwater, average of ten tests	0.916
Total air in feedwater from open heaters	0.931
	52 deg. fahr.
In solution in Croton water, average of three samples	4.325
	80 deg. fahr.
In solution in hot well water, average of four tests	0.269

7 These results show that Croton water gives off most of its dissolved air in passing through the open heaters, the amount of air occluded in the feedwater (0.931 per cent) being comparatively small. At 81 deg. fahr. and 2 in. of mercury absolute pressure when saturated with water vapor, this amount of air would occupy only 1/17 of the volume of the condensed steam.

8 The results obtained accord well with reports from other sources, although the author could find but little quantitative information on the subject. The fact was established, however, that very little air came into the condenser from the feedwater.

9 Having obtained the portions of air which enter the prime mover along with the steam, the next object of investigation was to determine the amount of air abstracted from the condenser by the air pump. Here again there were no precedents but it was determined to catch the air over water in a gasometer. For this purpose the discharge pipe of the air pump was piped to a

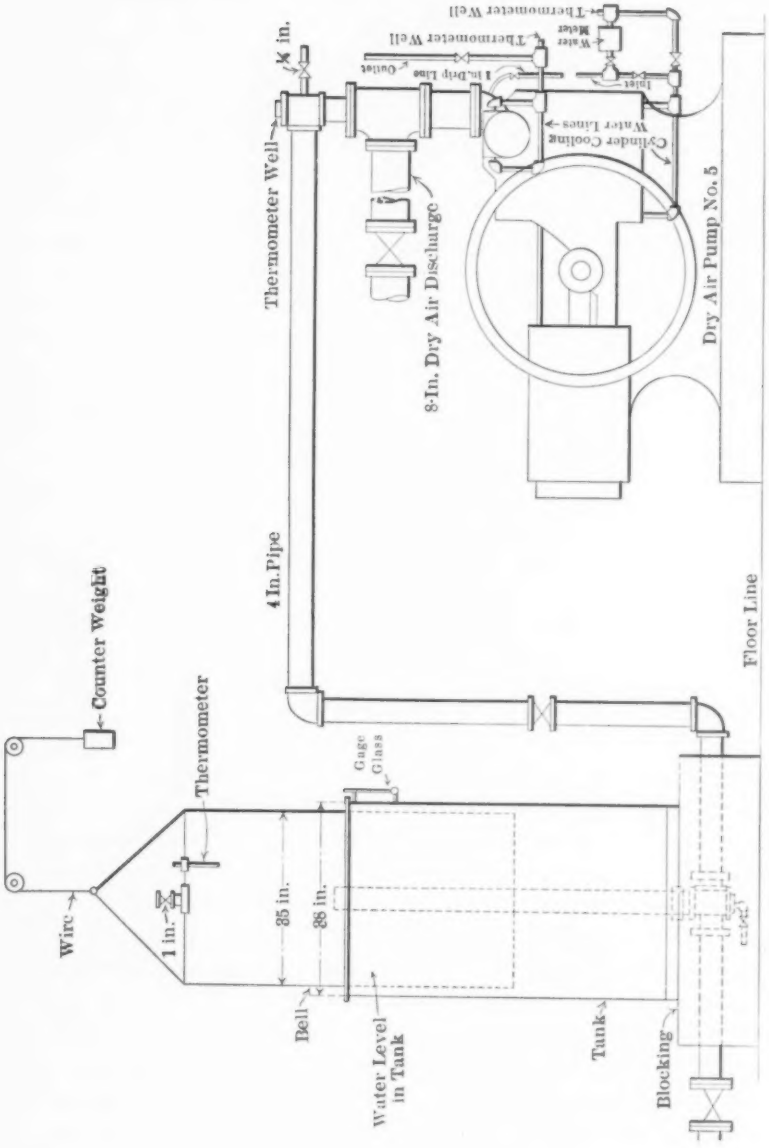

FIG. 2 LAYOUT FOR MEASURING AIR DISCHARGED BY DRY VACUUM PUMP

gasometer bell located in the basement in the vicinity of the air pump. A sketch plan of this outfit is shown in Fig. 2.

10 After the air pump had been running for a sufficient time to make sure that the conditions were constant, the valve was opened and the discharge from the air pump was allowed to enter the bell allowing it to rise. The time required to fill the bell was noted with a stop watch and from these data the amount of air delivered was calculated. The air in the bell was brought to atmospheric pressure and temperature before the final readings were taken.

11 Other tests were made with the pump shut off from the condenser to determine the air leakage in the pump itself and also with the suction piping blanked off at the condenser to determine the pipe leakage.

12 In a number of cases these readings showed excessive leakage in the condenser shell which was remedied before the final readings were taken. A second apparatus has been provided for Waterside No. 1 and both stations at the present time are so arranged that in a few hours the air leakage of all the air pumps as well as the condensers may be tested. This is a great help in operating and serves a useful purpose in enabling attention to be given where there was no indication formerly of leakage.

13 A great many air tests have been made on all of the units in the two stations. Usually readings were taken during the time necessary to fill the bell (from 1 to 20 minutes) of the air temperature in the bell, the air suction temperature, air discharge temperature, vacuum at the top of the condenser, vacuum in the air suction line to the pump, hot well temperature, circulating water inlet and outlet temperature, amount of cylinder cooling water with its inlet and outlet temperatures, revolutions per minute of the pump, power delivered by the prime mover and sometimes indicator cards on both air and steam cylinders. Table 1 shows a collection of some of the more interesting of these tests.

14 In the experimental apparatus used to obtain the data for the paper on The Transmission of Heat, it was impossible to detect any difference of temperature due to the segregation of air in the various parts of the single tube condenser. On the large condensers on which these air tests were made such

TABLE 1 RESULTS OF TESTS ON DRY AIR PUMPS FOR SURFACE CONDENSERS

Number of Tests Made	Date of Tests	Load on Turbine, Kw.	Vacuum at Top of Condenser Corrected to 30 In. Barometer	Vacuum of Pump Suction Corrected to 30 In. Barometer	Barometer	Hot Well	Circulating Water Inlet	Circulating Water Outlet	Air Suction	Air Discharge	Air in Bell	R.P.M. of Pump	Air discharged per Minute measured in Bell	Air discharged per Minute at Temperature of Air Discharge	I.H.P. Steam Cylinder	I.H.P. Air Cylinder	Mechanical Efficiency of Pump, Per Cent	Water Used to Cool Air Cylinder		
			Inches of Mercury			Temperatures, Deg. Fahr.							Cubic Feet					Temperature Inlet, Deg. Fahr.	Temperature Outlet, Deg. Fahr.	Cubic Feet used per Minute
3	2/14/11	6,500	28.49	28 61	30.09	54.7	35.7	48.8	63.0	144.7	94.3	70 0	12.94	17.15	26 92	14.80	57.1	....	46.3	....
5	2/14/11	8,200	28.43	28 61	30.07	55.0	33.0	49.8	68.0	141.7	93.3	61.4	8 59	11.08	22.85	10.36	45.7	....	52.6	....
5	2/14/11	10,400	28.16	28.40	30.04	63.0	33.0	54.2	81.2	139.4	93.1	60.4	5.82	7.40	21.52	10.24	47.6	....	53.7	....
5	2/18/11	6,266	28.50	28.60	29.92	53.0	33.6	47.9	62.8	138.2	96.5	67.2	12.65	15.65	22.51	11.94	53.0	38.56	45.0	....
5	2/18/11	8,133	28.46	28 60	29.92	57.8	33.4	51.1	73.4	136.2	96.0	66.4	6.86	8.50	21.80	9.35	42.9	38 00	45.6	2.21
5	2/18/11	10.132	28.11	28.34	29.92	68.0	33.0	54.2	84 9	137 0	95.5	65.4	5.75	7.12	20.08	10.85	58.9	38.00	46.0	2.19
3	2/18/11	10,000	28.08	28.32	29.90	68 3	33.3	54.7	85.2	144.8	97.3	74.0	5.93	7.83	23 08	12.30	53.3	37.20	43.0	3.91
2	2/18/11	10,000	27.95	28.27	29 90	69 0	34.0	56.0	86.0	144.5	96.3	72.5	5.89	7.78	23.89	11.75	49.2	38.30	49.8	....
2	2/18/11	10,000	27.90	28.23	29.90	69.5	34.5	56 0	87.7	147.3	96.0	72.0	5.27	7.07	23.32	10.58	45.3	40.00	67.8	0.49
*3	2/28/11	No load	....	29.30	30.20	....	....	....	....	99.3	98.6	75.6	1.31	....	25.32	13.99	55.3	37.70	43.0	3.14
4	3/14/11	6,075	....	29.00	30 39	....	....	....	55.0	114.0	83.5	77.0	8.28	....	....	....	....	....	....	....
6	3/14/11	9,252	....	28 99	30.35	....	....	....	57.0	111 2	88 2	75.5	6.94	....	....	....	....	....	....	....
4	3/14/11	11,613	....	28.93	30.31	....	....	....	69.8	118.9	85.5	75.5	6.33	....	....	....	....	....	....	....
*2	3/15/11	No load	....	29.10	29.98	....	....	....	....	85.7	96.5	76.5	0.63	....	....	....	....	....	....	....
	1/26/12	9,100	27.67	28.74	29.89	....	33	60	91.8	105.0	87.2	....	29.25	31.25	....	....	....	....	....	....
	1/26/12	5,970	28.15	28.65	30.03	....	33	51	91.0	100.5	88.0	....	29.60	30.95	....	....	....	....	....	....
6	4/24/11	3,022	....	28 05	30.25	....	....	....	59.1	184	95	81	39 5	....	....	....	....	....	....	....
4	4/24/11	5 204	....	28.00	30.18	....	....	....	60 6	198	98	80	38.5	....	....	....	....	....	....	....
6	4/20/11	8,258	....	28.10	29 78	....	....	....	59.0	164	93	85	36 8	....	....	....	....	....	....	....

*Valve in suction line closed.

a difference of pressure and temperature does exist, the differences of pressure being quite strongly marked. The curves for a number of these condensers, showing the increase in drop with the increase in amount of steam consumption, are illustrated in Fig. 3.

15 A number of experiments were made to find if this drop

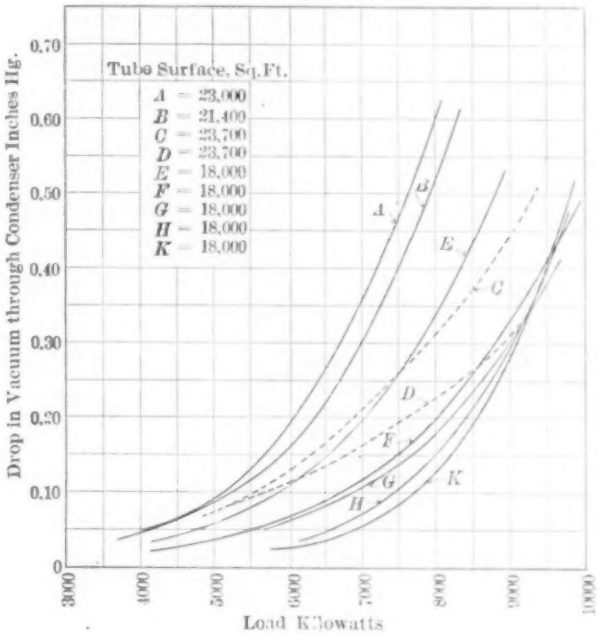

FIG. 3 CURVES SHOWING RELATION BETWEEN LOAD AND DROP THROUGH CONDENSER

was continuous through the tube surface or whether there were restrictions concentrating the drop in one or more places, and a number of experiments were made to reduce the amount of drop by taking out tubes thus opening channels into the body of the tube surface. The results here were apparently negative.

16 To ascertain the effect of known quantities of air on the vacuum suitable orifices of various sizes were installed at the top of the condenser so that the introduced air might mix with the incoming steam. These orifices were the ordinary circular orifices through thin plates and the amounts of air entering the condenser were checked up by the air delivered into the

air bell from the air pump. These results are shown in Fig. 4.

17 The tests in each case were run at constant load which was considerably below the capacity of the condenser.

18 Four sets of tests were run to determine the effect of load on air leakage, two on machines which were in bad condition and two on comparatively tight machines. The results are shown in Fig. 5. It should be noted that in every case the air leakage is less at the higher loads.

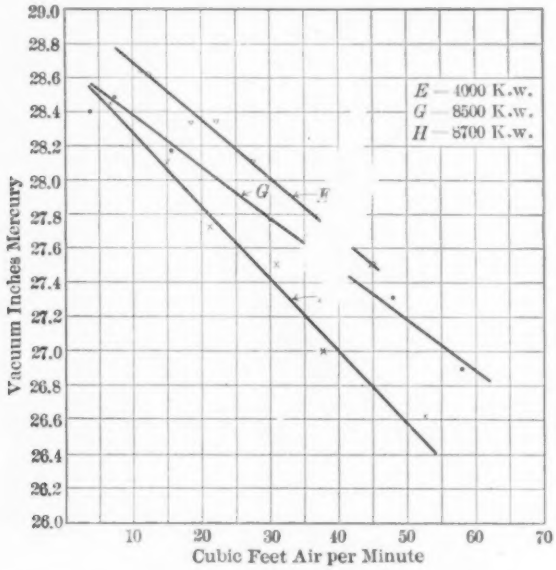

FIG. 4 CURVES SHOWING EFFECT OF AIR LEAKAGE AT CONSTANT LOAD

19 In the paper on The Transmission of Heat it was inferred from Smith's curves that the heat transfer varied as the fifth power of the ratio $\frac{p_s}{p_t}$. Since that time many tests have been run from which sufficient data to calculate this relation were obtained, and the results are shown in Fig. 6. None of the tests approach the capacity of the tube surface but the results are consistent.

CONCLUSIONS

20 Where open heaters are used very little air is carried to the prime mover with the steam. The volume of this air at at-

mospheric pressure and temperature will probably not exceed 1 per cent of the volume of the feedwater. As the whole system is under pressure from the pump nearly to the final stage of the turbine no air leakage can take place up to that point.

21 The air discharged by the dry air pump at atmospheric pressure and temperature from units between 5000 kw. and 20,000 kw. in size varies from 1 cu. ft. per min. when the units are in the

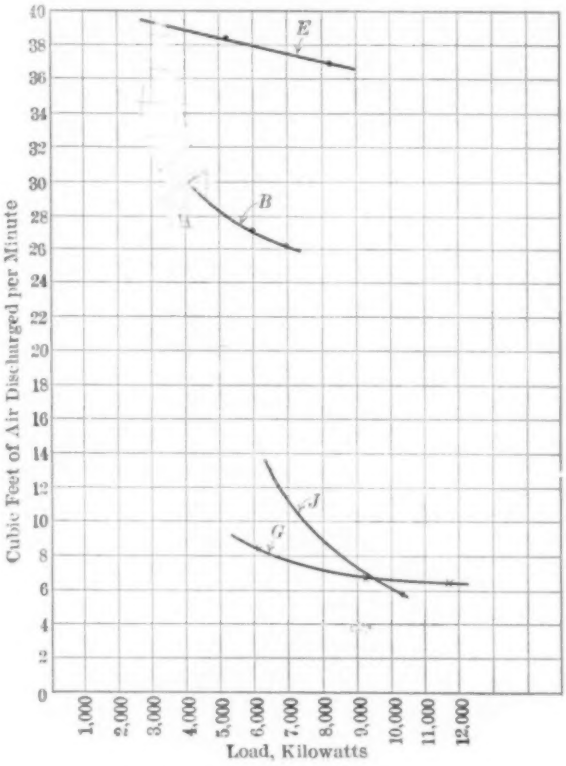

FIG. 5 CURVES SHOWING EFFECT OF LOAD UPON AIR LEAKAGE

best condition, to 15 or 20 cu. ft. where ordinary leakage is present, and to 30 or perhaps 40 or 50 cu. ft. where the units are in very bad condition.

22 Most of this leakage comes into the condenser and exhaust passages through minute leaks in the cast-iron shells, gaskets, and expansion joints. It is exceedingly difficult to detect these leaks with a candle flame but most of them may be detected by

filling the condenser with warm water under a slight head. That portion of the leakage occuring in the dry air pump and piping may be detected by the shut-off test on the pump. The volume of this leakage is larger than generally supposed and is much larger when the pump is warm.

23 The drop in vacuum through condensers of standard de-

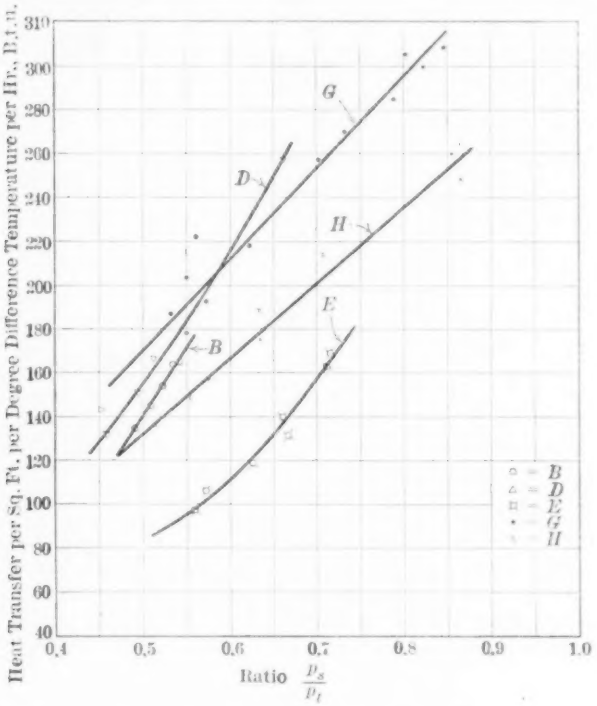

FIG. 6 CURVES SHOWING RELATION BETWEEN RATIO $\frac{p_s}{p_t}$ AND HEAT TRANSFER

sign is larger than it should be and a design which freely admits the steam to the tube surface should give better results. The opening of channels into the surface will not always improve conditions. The results indicate that wide shallow condensers should give better results although in at least two instances deep narrow condensers gave good results.

24 The falling off of vacuum with increased air leakage is considerable and the work necessary to maintain tight condensers more than pays for itself.

25 Air leakage decreases with an increase of load. This fact in some measure corrects the decrease of vacuum at the increased loads.

26 The exponent of the ratio $\frac{p_a}{p_t}$ on page 1161 of Volume 32 of the Transactions of the Society may be changed from 5 to 2 pending further experiments to establish a better value.

27 During these experiments in which four different kinds of dry air pumps were investigated it became evident that the volumetric efficiency of a pump working between ¼ lb. and 15 lb. absolute is very poor indeed. The difficulties of keeping valves, pistons, glands and packing tight against such a tenuous fluid as air at vacuum pressures are of much greater moment than is usually believed. It is hoped that some of the types of kinetic air pumps now being placed on the market will reduce these difficulties to a practical amount.

28 Considerable difficulty was experienced in securing vacuum temperatures during these experiments and most of the work has been repeated a number of times where such temperatures are essential. It has been impossible up to the present time to secure readings which were entirely consistent and the work along this line will be carried on until the cause of the discrepancy is ascertained and correct figures obtained.

29 The author wishes to acknowledge the efficient aid rendered him in conducting these experiments by his assistants, Messrs. E. B. Rickets, J. H. Lawrence and J. S. Kerins, and by P. E. Reynolds, engineer of tests.

MEASUREMENT OF NATURAL GAS

By Thos. R. Weymouth

ABSTRACT OF PAPER

In the measurement of natural gas there are two distinct requirements to be met. The first relates to the determination of the open flow capacity of gas wells, for which two principal methods are used, namely, the "minute pressure" measurement and the pitot tube. This paper describes these methods, giving the proper formulae to be used with them.

The second requirement is that of the measurement of flow in closed pipes. While large proportional meters are used extensively for this purpose, the high pressures often encountered have a tendency to introduce serious errors, making it necessary to resort to more accurate devices. Of these, the best known are described, namely, the Thomas electric meter, the venturi meter, the pitot tube, and the orifice. The general formulae are given for the last three named, together with their derivations.

MEASUREMENT OF NATURAL GAS

By Thos. R. Weymouth, Oil City, Pa.

Member of the Society

One of the most important problems encountered in natural gas engineering is the accurate measurement of gas in large volumes. For the purpose of discussion, the subject can be divided into two general classifications, namely, open flow and closed flow measurements. The first relates principally to the determination of the volume of gas a well is capable of producing, while the second refers to all closed pipe flows.

2 Under the first head, there are two general methods in use for measuring the open flow of a well, that is, the quantity of gas the well is capable of producing against an absolute back pressure of one atmosphere. The first of these is purely approximate, and is called the "minute pressure" measurement, wherein the well is permitted to blow into the air until its "head" is blown off, and is then suddenly closed in by a quick-acting gate, the accumulated pressure being noted at the end of one minute from the time the gate is closed. Then, Q, the open flow capacity of the well in cubic feet per day, on an atmospheric pressure base of 14.4 lb. per sq. in. absolute, becomes

$$Q = 1440\ V\left\{\frac{p_m + P_o}{P_o} - \frac{P_o}{P_o}\right\} = 1440\ V\ \frac{p_m}{P_o} = 100\ V p_m \quad \ldots\ldots\ldots [1]$$

in which

V = volume of well tubing in cu. ft.

p_m = minute pressure in lb. per sq. in. gage

P_o = atmospheric pressure in lb. per sq. in. absolute

3 This equation assumes that while the gas is flowing freely into the air, the well is filled with it at a uniform pressure of one atmosphere, and at the end of one minute it is filled with gas at a

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uniform pressure equal to the minute pressure. Neither of these assumptions is correct, since in both cases there is a gradually increasing pressure from the top of the well towards the bottom, similar to that in a pipe line.[1] The resulting errors tend to counteract each other, however, and if care is taken in making the measurements a comparative idea will be obtained of the ability of the well to produce gas against a back pressure equal to a mean between the atmosphere and the minute pressure. Since the open flow will be greater than the flow against this back pressure, the assumption that the minute pressure gives an open flow measurement errs on the side of conservatism.

4 The second method of open flow measurement is that by means of the pitot tube, and is the one generally used for accurate determinations of well capacities. This method has been described in detail by Prof. S. W. Robinson,[2] who was probably the first to use it in connection with gas. The method was further discussed by him in the Geological Reports of Ohio,[3] wherein tables are published by means of which the flow may be computed from the pitot tube observations. These tables have been in general use by natural gas men for many years, and are based on the formula

$$Q = 1{,}462{,}250\, d^2 \left\{ \left(\frac{P_1}{P_o} \right)^{0.29} - 1 \right\}^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots [2]$$

5 Equation [2] was derived from the formula for adiabatic flow

$$v^2 = \frac{2gn \times 144\, P_o}{(n-1)\,\delta} \left\{ \left(\frac{P_1}{P_o} \right)^{\frac{n-1}{n}} - 1 \right\} \quad \ldots\ldots\ldots\ldots [3]$$

In equations [2] and [3]

v = velocity of flowing gas, ft. per second
P_o = absolute pressure of atmosphere, lb. per sq. in.
n = ratio of specific heats
δ = weight per cu. ft. of gas at pressure P_o
G = specific gravity of gas, air = 1.0
P_1 = absolute pressure shown by pitot tube, lb. per sq. in.
d = internal diameter of well mouth, in.
Q = open flow capacity of well, cu. ft. per 24 hr.

6 In computing his tables, Professor Robinson used the following values for these quantities:

[1] Problems in Natural Gas Engineering, The Journal, Am. Soc. M. E., May 1912, p. 723, Fig. 5.

[2] Van Nostrand's Engineering Magazine, August 1886.

[3] Vol. 6, 1888.

$$n = 1.408$$
$$2g = 64.3$$
$$P_o = 14.6$$
$$G = 0.6$$
$$\delta = 0.0807 \frac{T_o}{T} G$$
$$T = T_o = T_s = 492 \text{ deg.}$$

where

T_o = absolute temperature of melting ice
T = absolute temperature of flowing gas
T_s = absolute temperature of storage

all being in fahrenheit degrees.

7 The ratio of specific heats for natural gas is more nearly equal to 1.266 than to the value used above. Inserting this value and using values for the other quantities as follows:

$$T = 500 \text{ deg.}$$
$$T_s = 520 \text{ deg.}$$
$$P_o = 14.4$$

Storage Pressure, $P_s = 14.65$

the formula becomes

$$Q = 1{,}758{,}560\, d^2 \left\{ \left(\frac{P_1}{14.4} \right)^{0.21} - 1 \right\}^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

8 At 60 lb. gage pressure per sq. in. (the highest given by Professor Robinson), formula [4] makes the flow about 1 per cent lower than that derived from formula [2], whereas at 1 lb. it gives a result about 4 per cent higher.

9 The open flow measurement is not required with the same degree of accuracy as that of closed flow, where gas is being bought and sold, and at best is only intended to give a fair idea of the production. Even if an accurate measurement were made it would give no precise notion of the real productive value of the well when it must feed into the line against a high back pressure. Consequently, if it is desired to know the capacity under these conditions it is best to insert a pitot tube into the lead line from the well while operating under normal conditions. This involves the second division of the subject, namely, closed flow methods of measurement.

10 In closed flow measurements of gas where the volumes are small and the pressures are low, the ordinary positive gas meters are satisfactory, but for wholesale measurement some other type of meter must be adopted because of the prohibitive size of the positive type necessitated by large flows. The matter is further complicated

by the fact that usually when gas is bought or sold in large quantities it must be measured under varying pressures, necessitating the use of a correction factor to reduce the meter readings to quantities on a standard pressure base. There are many different types of proportional meters in use for this purpose, equipped with recording gages. The meter registration is then multiplied by a factor dependent upon the record of the gage in order to correct it to volumes based on the proper buying or selling pressure.

11 In addition to proportional meters there are a number of other devices in use which aim to attain a higher degree of accuracy, among which the most prominent are the venturi meter, the Thomas electric meter, the orifice, and the pitot tube.

12 The venturi meter is used principally for low pressure measurements, and while extremely accurate, is handicapped by possessing a complicated formula for the expression of volumes, which renders very tedious the computation of results from its records, and which demands a knowledge of the ratio of specific heats of the gas at constant pressure and constant volume, respectively. This formula is as follows: [1]

$$Q = 210{,}840\, A_2 \frac{T_s}{P_s} \sqrt{\frac{n}{(n-1)\,G}} \frac{P_1}{\sqrt{T_1}} \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \sqrt{\frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{1}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}}} \quad \ldots\ldots [5]$$

wherein

Q = flow, cu. ft. per hr., on absolute pressure and temperature bases P_s and T_s
n = ratio of specific heats
G = specific gravity of gas
A_1 = area of venturi meter at entrance, in sq. ft.
A_2 = area of venturi meter at throat, in sq. ft.
P_1 = absolute pressure at entrance, lb. per sq. in.
P_2 = absolute pressure at throat, lb. per sq. in.
T_1 = absolute temperature at entrance, deg. fahr.
T_2 = absolute temperature at throat, deg. fahr.

13 For any particular meter and quality of gas, and assuming definite values of P_s, T_s, and T_1, equation [5] reduces to the simpler form

$$Q = KP_1R^{\alpha}\sqrt{\frac{1-R^{\beta}}{1-CR^{2\alpha}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

[1] See Appendix No. 1.

where

$$K = 210{,}840\, A_2 \frac{T_s}{P_s} \sqrt{\frac{n}{n-1} \frac{1}{GT_1}}$$

$$C = \left(\frac{A_2}{A_1}\right)^2$$

$$R = \frac{P_2}{P_1}$$

$$a = \frac{1}{n}$$

$$\beta = \frac{n-1}{n}$$

14 The Thomas electric meter [1] is an extremely accurate instrument for the measurement of gas and is comparatively simple in construction. It operates on the principle that the energy required to heat any quantity of a given gas through a definite temperature range is directly proportional to the weight of the gas heated, and can therefore be used as a measure of it. In practice, electrical energy is used to supply the heat and is automatically controlled in such manner as to cause the temperature increase of the flowing gas to remain fixed at a value of about 2 deg. fahr. The electrical energy required to do this is measured by means of wattmeters, the dials of which can be graduated to read direct in cubic feet of gas based on any desired standards of pressure and temperature. The method is independent of the actual temperature and pressure of the gas entering the meter, depending solely upon its specific heat at constant pressure. Since the records of the device are obtained through the medium of wattmeters, they can therefore be made to give either a rate reading of flow, an integrated or totalized reading, or both.

15 This meter has been subjected to very thorough tests in conjunction with the pitot tube and venturi meter [1] which have firmly established the accuracy and reliability of all three of the devices.

16 The Thomas meter requires a supply of electric energy equivalent to about 1 kw-hr. for each 75,000 cu. ft. of free gas. This fact greatly restricts its use in the measurement of natural gas, especially in the field where such power is not usually available.

17 The most practical devices known to the author for general use in the accurate measurement of gas are the pitot tube and the

[1] Measurement of Gases, Carl C. Thomas, The Journal of the Franklin Institute, November 1911.

orifice, both of which are extremely simple in construction and are based on fundamentally correct principles. The formula expressing the flow of gas is identical for the two instruments for the range through which they are usually used.

18 The pitot tube was first described before the French Academy of Sciences in 1732 by its inventor, Pitot, who stated that it followed the principle of the law of falling bodies, namely $v^2 = 2\ gh$. For over a century after its discovery this wonderfully simple and accurate device was regarded merely as a scientific toy, until about 1850, when D'Arcy adapted it to closed pipe flows by the addition to the bent tube of a static pressure tube. It was then found that the difference in height of liquid in the two columns of a U-gage connected to the dynamic and static tips of the tube respectively, gave a true measure of the velocity, and followed Pitot's originally stated principle exactly, viz., $v^2 = 2\ gh$.

19 Prof. S. W. Robinson seems to have been the first to have applied the pitot tube to the measurement of gases. In 1873 he experimented with it in determining the velocity and form of jets of air from orifices, which led to the development of a commercial pitot tube for gas measurement.

20 If a pitot tube is inserted in a pipe conveying gas and a traverse made over the cross-section of the pipe, it will be found that the highest velocity exists at the center and the lowest at the walls. For this reason it has been customary for experimenters to determine the particular point in the cross-section of the pipe where the velocity corresponds to the mean velocity of the whole cross-section, and to keep the tube at that point for the determination of the quantity of flow as computed from the velocity readings. This point varies according to the condition of roughness of the interior surface of the pipe in which the tube is inserted, and accordingly it is necessary to determine it for each particular pipe used.

21 When a tube is to be used as a permanent gas meter, it is better, because of the fact just noted, to construct the pipe with extreme care, having the interior of uniform size and highly polished, and then to place the tip in the exact center of it. This necessitates the inclusion in the theoretical formula of a factor which in this paper is termed the "efficiency" of the tube, in order to reduce the flow, as indicated by the tube readings, to the value it would assume were the tube at the point corresponding to mean velocity instead of being at that of maximum velocity.

22 The tubes used by the author are all constructed in this man-

ner. The instrument as a whole is termed a pitot tube (comprising both the pipe and the bent tube inserted in it), and the bent tube portion of it is designated as the "tip." The pipe consists of 12 ft. of seamless drawn brass tubing with highly polished interior surface and gaged to accurate size. About 27 in. from the outflow end a brass saddle is sweated on, through which the tip projects into the tube, and which also contains the static pressure connection, which is merely a hole drilled through the saddle and tube flush with the inside wall. The tip is constructed in such manner as to be readily removable for cleaning, and yet exactly central in the tube when re-inserted.

23 A series of standard tubes of this type are maintained, against which all new tubes are calibrated. These standards, ranging in size from 2 in. to 6 in., were themselves calibrated by exhaustive tests against a large gas holder under different conditions of flow and pressure. The coefficients thus obtained were found to be constant throughout the complete range of water readings under which the tubes are used and also for all pressures.

24 The accuracy of these standards is attested by the paper by Professor Thomas already referred to, in which are given the results of a 45-day test at the Brave pump station of the People's Natural Gas Company of Pittsburgh, during which the Thomas electric meter was connected in series with pitot tubes that had been calibrated against the standards above referred to. A total of about 337,000,000 cu. ft. of gas was measured with a difference between the two meter records of 0.2 per cent.

25 The general formula [1] of the pitot tube and orifice is expressed by the equation

$$Q = 218.44\ Ed^2 \frac{T_s}{P_s} \sqrt{\frac{hP}{TG}} \qquad [7]$$

where

E = efficiency of the tube, expressed as a decimal
d = internal diameter of the tube, in.
T_s = absolute temperature of measurement base, deg. fahr.
P_s = absolute pressure of measurement base, lb. per sq. in.
G = specific gravity of gas —air = 1.0
P = absolute static pressure of flowing gas, lb. per sq. in.
T = absolute temperature of flowing gas, deg. fahr.
h = velocity head of flowing gas, in in. of water

[1] See Appendix No. 2.

Assuming standard conditions of storage temperature and pressure, namely, $P_s = 14.65$ lb. per sq. in. absolute, and $T_s = 60 + 460 = 520$ deg. absolute, equation [7] becomes

$$Q = 7753\ Ed^2 \sqrt{\frac{hP}{TG}} \quad \ldots\ldots [8]$$

26 For any particular installation the gravity of the gas is once determined, and, with an assumed value for flowing temperature, is included with E and d to form a coefficient, making the formula read

$$Q = C\sqrt{hP} \quad \ldots\ldots [9]$$

where

$$C = 218.44\ Ed^2 \frac{T_s}{P_s\sqrt{TG}} \quad \ldots\ldots [10]$$

27 If the flowing temperature is other than the value assumed in making up the coefficient, the flow is corrected by multiplying by the factor $\sqrt{\frac{T_a}{T}}$, where T_a is the assumed flowing temperature.

28 The value of E is remarkably constant for tubes ranging in size from 2 in. to 5 in., as will be seen from the following tabulation, taken from the standard tubes already referred to:

Actual Diameter of tube, d	Efficiency, E
3.0625	0.8534
3.96875	0.8515
5.0469	0.8541
Mean	0.8530

29 Assuming the average value of E to be 0.8530, the coefficient of any pitot tube constructed as above described will be

$$C = 381.7d^2 \quad \ldots\ldots [11]$$

based on a storage pressure of 14.65 lb. absolute, storage temperature of 60 deg. fahr., flowing temperature of 40 deg. fahr., and specific gravity of 0.600.

30 In using the pitot tube it is essential that a straight run of at least 30 ft. of pipe of the same diameter as the tube be installed ahead of it, in order to avoid errors due to eddies in the flow. It is also necessary to have absolutely no leaks in the pressure lines leading from the tube to the differential gage.

31 In practice the U-gage is modified by making one leg to consist of a cast-iron water pot having a cross-sectional area many times

greater than that of the glass tube comprising the other leg of the U. Thus the height of the water level in the pot does not change appreciably for a change in the glass tube from zero to full scale reading of about 28 in. The pot is provided with a gage glass to show the level of the water in it, to which the zero of the adjustable scale is set, thus necessitating the reading of but one leg of the U-gage, that comprised by the long gage glass.

32 The scales used in the tubes described are graduated to 28 in., and are used under all conditions of pressure. If the static pressure of the flowing gas is equal to that of the atmosphere, the full scale water reading corresponds to a velocity in the pipe of over 22,000 ft. per minute. As the static pressure increases, the velocity equivalent becomes smaller and at 100 lb. gage is about 8000 ft. per minute. Thus, the higher the pressure under which the tube operates, the greater will be the range of water pressure permissible to use.

33 It is not considered advisable to use the pitot tube for flows giving a differential pressure less than 2 in. of water, owing to the high percentage of error resulting from any personal error in reading the gage at such low readings. With a maximum scale reading of 28 in. this means that under atmospheric pressure conditions the ratio of maximum to minimum quantities of gas susceptible of accurate measurement by the pitot tube is about 3.75 to 1. Assuming the same permissible velocity (22,400 ft. per minute) under a pressure of 100 lb. gage, the maximum water scale reading could be increased to more than 200 in., giving the tube a range of 10 to 1. The author is now conducting experiments along this line, but as yet has not progressed far enough with the work to state authoritatively to what extent this range may be extended without affecting the constancy of the coefficient.

34 At present the usual method of obtaining the required range of capacity in a pitot tube measuring station is to install the tubes in batteries so that more or less of them may be cut in or out of service, as conditions demand, for obtaining accurate readings.

35 In accordance with Torricelli's theorem, the velocity of issue of a fluid from an orifice follows the law of falling bodies, and the formula deduced therefrom expressing the quantity of flow is identical with that of the pitot tube. The vena contracta of the jet, however, causes the coefficient of flow (termed "efficiency" herein) to have a lower value than the corresponding factor in the pitot tube formula. For differential pressures greater than about 4 in. of water

it is sensibly constant, and thus the orifice furnishes a simple means of gas measurement that compares favorably with the pitot tube.

36 Instead of a water column, requiring personal observation at regular stated intervals, a recording differential gage may be used, constructed with the working parts enclosed in a casing to which the static pressure is applied, while the dynamic pressure is applied to the interior of the gage. Such an instrument has been used with success in connection with an ordinary recording pressure gage for determining the static pressure, thus obviating the necessity of having an attendant constantly on duty to make observations.

37 The simplicity of the pitot tube and orifice, both in manipulation and computation, and the high degree of accuracy attainable with them, render them invaluable in the measurement of natural gas in large volumes and at high pressures. The tube has been in commercial use for many years, with highly satisfactory results. The orifice, although one of the oldest known measuring devices, has not until recently been used extensively in the measurement of gas under high pressure closed flow conditions. It is being rapidly developed, however, and bids fair to come into extensive use as a simple, accurate gas-measuring device.

38 The only disadvantage in the use of these two instruments is the necessity for the computation of results. Their largest field has consequently been in the measurement of sufficient quantities of gas to warrant the maintenance of stations with observers constantly in attendance. The adoption of the recording gages, however, has greatly extended their usefulness and has considerably simplified the problem of the accurate measurement of gas under pressure.

APPENDIX NO. 1

39 The general formula of the venturi meter is derived on the assumption that the change in state of the gas between entrance and throat of the meter takes place adiabatically. Hence the drop in potential energy, as evidenced by the respective pressures is equal to the increase in kinetic energy indicated by the respective velocities at entrance and throat.

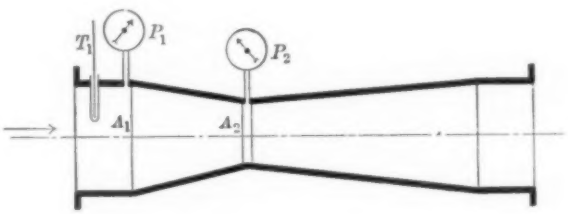

FIG. 1 DIAGRAM OF VENTURI METER

40 In Fig. 1, which diagrammatically represents a venturi meter, let

A_1 = area of cross-section at entrance, sq. ft.
A_2 = area of cross-section at throat, sq. ft.
P_1 = absolute pressure, lb. per sq. in. at entrance
P_2 = absolute pressure, lb. per sq. in. at throat
v_1 = absolute velocity at entrance, ft. per second
v_2 = absolute velocity at throat, ft. per second
W = weight of gas passing per second
δ_1 = density of gas at entrance, lb. per cu. ft.
δ_2 = density of gas at throat, lb. per cu. ft.
T_1 = absolute temperature of gas at entrance
T_2 = absolute temperature of gas at throat
T_s = absolute temperature of storage base, deg. fahr.
P_s = absolute pressure of storage base, lb. per. sq. in.
G = specific gravity of gas, air = 1.0
T_o = absolute temperature of 32 deg. fahr. = 492
P_o = absolute pressure of atmosphere, lb. per sq. in.
δ_w = weight of water per cu. ft. at 60 deg. fahr. = 62.37 lb.
δ_a = density of air at T_o and P_o
δ_{a1} = density of air at T_1 and P_1

Kinetic energy of gas at entrance $= \dfrac{Wv_1^2}{2g}$

Kinetic energy of gas at throat $= \dfrac{Wv_2^2}{2g}$

Increase in kinetic energy is

$$E_k = \frac{Wv_2^2}{2g} - \frac{Wv_1^2}{2g} = \frac{W}{2g}(v_2^2 - v_1^2) \quad \ldots\ldots [12]$$

41 The work done on one pound of gas in expanding adiabatically from P_1 to P_2 is

$$\frac{144\ (P_1V_1 - P_2V_2)\ n}{n-1}$$

and the change in potential energy in W pounds of gas per second is

$$E_p = \frac{144\ W\ (P_1V_1 - P_2V_2)\ n}{n-1} \quad \ldots\ldots [13]$$

42 Hence

$$\frac{W}{2g}(v_2^2 - v_1^2) = \frac{144\ Wn}{n-1}(P_1V_1 - P_2V_2) = \frac{144\ Wn}{n-1}\left(\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2}\right)$$

$$v_2^2 - v_1^2 = \frac{288\ gn}{n-1}\left(\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2}\right) \quad \ldots\ldots [14]$$

But $v_1A_1\delta_1 = v_2A_2\delta_2$ and $P_1\delta_2^n = P_2\delta_1^n$ or $\dfrac{\delta_2}{\delta_1} = \left(\dfrac{P_2}{P_1}\right)^{\frac{1}{n}}$. Hence

$$v_1 = v_2\frac{A_2\delta_2}{A_1\delta_1} = v_2\frac{A_2}{A_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$v_2^2 - v_1^2 = v_2^2 - v_2^2\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} = \frac{288\ gn}{n-1}\left(\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2}\right)$$

$$v_2^2 = \frac{\dfrac{288\ gn}{n-1}\left(\dfrac{P_1}{\delta_1} - \dfrac{P_2}{\delta_2}\right)}{1 - \left(\dfrac{A_2}{A_1}\right)^2\left(\dfrac{P_2}{P_1}\right)^{\frac{2}{n}}} \quad \ldots\ldots [15]$$

43 But $\dfrac{P_1}{\delta_{s1}T_1} = \dfrac{P_o}{\delta_s T_o}$ and $\delta_{s1} = \delta_s\dfrac{P_1T_o}{P_oT_1}$

$$\delta_1 = G\delta_{s1} = G\delta_s\frac{P_1T_o}{P_oT_1}$$

Also

$$\frac{P_1}{\delta_1^n} = \frac{P_2}{\delta_2^n} \quad \therefore \quad \delta_2 = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\delta_1 = G\delta_s\frac{P_1T_o}{P_oT_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

Hence

$$\frac{P_1}{\delta_1} - \frac{P_2}{\delta_2} = \frac{P_1}{G\delta_s\dfrac{P_1T_o}{P_oT_1}} - \frac{P_2}{G\delta_s\dfrac{P_1T_o}{P_oT_1}\left(\dfrac{P_2}{P_1}\right)^{\frac{1}{n}}} = \frac{P_oT_1}{G\delta_sT_o}\left\{1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\} \quad \ldots\ldots [16]$$

Substituting in [15], and extracting the square root

$$v_2 = \sqrt{\frac{\frac{288\,gn}{n-1}\frac{P_o T_1}{G\delta_a T_o}\left\{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right\}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [17]$$

44 The flow, in cubic feet per hour, based on P_a and T_a is

$$Q = 3600\, A_2 v_2 \frac{P_2}{P_a}\frac{T_a}{T_2}$$

But

$$\frac{P_2}{T_2} = \frac{P_1\delta_2}{T_1\delta_1} = \frac{P_1}{T_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

Hence

$$Q = 3600\, A_2 \frac{T_a}{P_a}\frac{P_1}{T_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} v_2$$

$$Q = 210{,}840\, A_2 \frac{T_a}{P_a}\sqrt{\frac{n}{(n-1)\,G}}\frac{P_1}{\sqrt{T_1}}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\sqrt{\frac{1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}}{1-\left(\frac{A_2}{A_1}\right)^2\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}}} \quad \ldots\ldots [18]$$

APPENDIX NO. 2

45 The general formula for the pitot tube and the orifice is derived from the law of falling bodies. Let

T = absolute temperature of flowing gas, deg. fahr.
P = absolute pressure of flowing gas, lb. per sq. in.
δ = weight per cu. ft. of flowing gas, at P and T
G = specific gravity of flowing gas, air = 1.0
v = actual velocity of flowing gas, ft. per second
h_f = height in ft. of homogeneous column of gas at P and T producing v
h = corresponding height of water column in in.
δ_w = weight per cu. ft. of water = 62.37 lb. at 60 deg. fahr.
P_s = absolute storage pressure base, lb. per sq. in.
T_s = absolute storage temperature base, deg. fahr.
δ_a = weight per cu. ft. air at 32 deg. and 14.7 lb. = 0.08073 lb.
d = actual inside diameter of pipe or orifice, in.
E = efficiency of pitot tube or orifice
Q = flow in cu. ft. per hour at P_s and T_s

Then

$$v = \sqrt{2gh_f} = \sqrt{2g\frac{h}{12}\frac{\delta_w}{\delta}}$$

But

$$\delta = \delta_a G \frac{P}{14.7}\frac{492}{T}$$

Hence

$$v = \sqrt{2g\frac{h}{12}\frac{\delta_w}{\delta_a}\frac{14.7}{492}\frac{T}{PG}} \quad \ldots\ldots [19]$$

$$Q = 3600\frac{\pi d^2}{4 \times 144} v \frac{P}{P_s}\frac{T_s}{T}E$$

$$Q = 218.44\ Ed^2\frac{T_s}{P_s}\sqrt{\frac{hP}{TG}} \quad \ldots\ldots [20]$$

THE CENTRIFUGAL BLOWER FOR HIGH PRESSURES

By Henry F. Schmidt

ABSTRACT OF PAPER

In this paper the author gives the essential elements of blower design in convenient form for reference. An equation is derived representing the work done in a centrifugal blower and the differences between this type and a piston compressor are discussed. There is a discussion of efficiencies and characteristics of impellers in series, of guide-vane and volute types of blowers and variations of pressure in the case of guide-vane blowers as the volume of discharge increases. The theory of diffusion vanes is examined in detail, and an equation for the loss of kinetic energy derived.

Design of volute blowers is taken up and discussion of the influence of the relation between efficiency curve and discharge rate, as well as between the blower characteristic and efficiency curve in this type of blower; an equation for the velocity conversion efficiency is derived, and the constancy of pressure in a blower having a free vorte mathematically demonstrated. The Westinghouse design of volute blower is used for illustration.

The latter part of the article is devoted to methods of testing and calculation of centrifugal blowers.

THE CENTRIFUGAL BLOWER FOR HIGH PRESSURES

By Henry F. Schmidt,[1] East Pittsburgh, Pa.

Non-Member

In presenting the following notes, the writer does not claim originality, but offers the material simply as a collection of the essential elements of blower design in convenient form for reference.

2 The centrifugal blower for high pressures has suffered through the same lack of knowledge in regard to its possibilities as that which affected the introduction of the centrifugal pump for high pressures and high shaft speeds. A general impression was created among engineers from early experiments that centrifugal pumps and blowers were adapted for handling only large volumes against low heads and with comparatively low efficiency. This, however, has been shown to be an incorrect statement in the case of centrifugal pumps and it is also untrue in respect to centrifugal blowers.

3 With proper methods of construction of the rotating parts and due regard to the theory involved, it is possible to build blowers for practically any capacity and pressure with efficiencies ranging from 65 per cent to 80 per cent or more, the higher efficiency, of course, being for the larger volumes handled.

4 Before an intelligent conception of the characteristics of centrifugal blowers can be obtained it is necessary to examine the theory, which is not only simple but extremely interesting.

5 The first problem to be considered is the pressure created within the impeller due to the centrifugal force acting on the volume of air between the blades of the fan. This can be done as follows:

[1] Westinghouse Machine Company.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

6 In Fig. 1, imagine a tube extending radially from the shaft to the circumference, and assume the area of this tube to be A sq. ft. If an infinitely thin lamina of this tube be taken, the weight of the air contained in it will be $\rho A dr$ where ρ is the density of the air at that point. If this lamina is taken at radius r, then the centrifugal force acting on this small mass of air will be

$$dF = \frac{\rho A dr \mu^2}{gr} = \frac{\rho A\ (2\pi)^2\ rN^2 dr}{g}$$

in which N = revolutions per second. If the compression in the blades is adiabatic, the volume of a pound of air at any point is found from the relation $pv^\gamma = p_1 v_1{}^\gamma$ or

$$v = v_1 \left(\frac{p_1}{p}\right)^{\frac{1}{\gamma}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

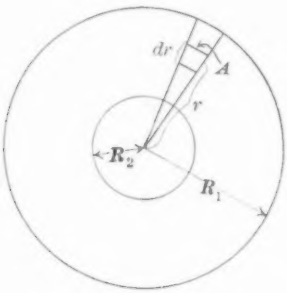

FIG. 1 PRESSURE WITHIN THE IMPELLER

But since the density is inversely proportional to the volume

$$\frac{1}{\rho} = \frac{1}{\rho_1}\left(\frac{p_1}{p}\right)^{\frac{1}{\gamma}} \quad \text{or} \quad \rho = \rho_1 \left(\frac{p_1}{p}\right)^{-\frac{1}{\gamma}}$$

7 Now also dF must equal the difference in pressure between the two sides of the lamina (dr) or $dF = A dp$. Hence, substituting the value of dF and ρ

$$dF = A dp = \frac{\rho_1 \left(\frac{p_1}{p}\right)^{-\frac{1}{\gamma}} A\ (2\pi)^2\ N^2 r dr}{g}$$

Transposing $\rho_1 \left(\frac{p_1}{p}\right)^{-\frac{1}{\gamma}}$

$$\frac{1}{\rho} p_1{}^{\frac{1}{\gamma}} p^{-\frac{1}{\gamma}} dp = \frac{(2\pi)^2\ N^2 r dr}{g}$$

Integrating between $\gamma = 0$ and $\gamma = R$, and p_1 and p_2,

$$v_1 p_1^{\frac{1}{\gamma}} \int_{p_2}^{p_1} p^{-\frac{1}{\gamma}} dp = \frac{(2\pi)^2 N^2}{g} \int_0^R r dr$$

which gives

$$v_1 p_1^{\frac{1}{\gamma}} \frac{\gamma}{\gamma - 1} \left[p_1^{\frac{\gamma-1}{\gamma}} - p_2^{\frac{\gamma-1}{\gamma}} \right] = \frac{(2\pi)^2 N^2 R^2}{2g} \quad \ldots\ldots\ldots [2]$$

8 But the first term of this equation is the work done in adiabatic compression or expansion, and the second term equals $\frac{\mu_1^2}{2g}$, hence the work done in the impeller is equal to that required to accelerate the air from rest to the velocity of the blade tips. Since, however, if the radial velocity is very small, the final tangential velocity of the air is μ_1 it is evident that one-half of the total work done on the air by the blades will be expended in accelerating it to a velocity equal to that of the blade tips, and the other half in compressing it to a pressure p_2. Hence, unless the final velocity of the air is reduced by converting the kinetic into potential energy, the maximum efficiency of a centrifugal compressor, neglecting all losses, can be but 50 per cent. It is evident, therefore, that some efficient means must be provided to convert the kinetic energy represented by the final velocity into potential energy represented by an increase in pressure. This can be done in one of three ways:

a By means of diffusion vanes placed around the periphery of the impeller

b By the use of a volute of proper design

c By a diffusion tube or inverted nozzle attached to the outlet of the compressor

9 The first of these methods is probably the most efficient at approximately the designed capacity if properly constructed, but is inefficient at large and small discharge rates. The additional cost of construction is also one of its disadvantages.

10 Theoretically and practically, the second method is the most efficient over a wide range of loads and has, if well designed, an efficiency of 65 per cent to 90 per cent. It is, however, more costly to construct than the third method.

11 Diffusion tubes attached to the discharge, though the cheapest to construct, have only cheapness and compactness to recommend them and cannot be used on the intermediate stages of multi-stage compressors. The reason for the inefficiency of diffusion tubes is that serious eddies are formed in them due to the varying pressures

at different sections of the discharge opening. The latter results from the fact that the air farthest from the shaft is compressed the most and has a higher velocity, which causes eddies.

12 From an examination of equation [2] representing the increase of pressure or work in foot-pounds per pound of air in the impeller, it is evident that the most peculiar characteristic of the centrifugal blower is that for a fixed number of revolutions per minute, or tip speed of the impeller, the work done remains constant so long as the volume entering the blower remains constant. It will

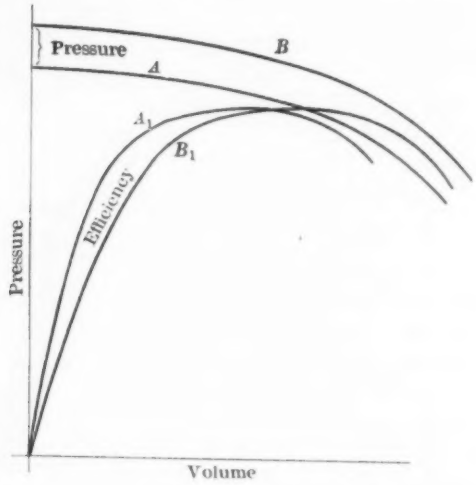

FIG. 2 PRESSURE AND EFFICIENCY CURVES, CENTRIFUGAL BLOWER

be noted that in this respect a centrifugal blower departs entirely from the piston compressor in that the volume discharged depends upon the pressure against which the blower is working, whereas in the piston blower for a fixed number of revolutions per minute the volume remains practically constant, the pressure of discharge varying according to the resistance placed in the discharge.

13 It will be noted that the work done in a centrifugal blower varies directly as the square of the tip speed, or if the diameter of the impellers is fixed, directly as the number of revolutions per minute and the capacity almost directly as the number of revolutions per minute. The result is that the pressure and efficiency curves for a change in the number of revolutions per minute vary somewhat from the manner shown diagrammatically in Fig. 2. The point to

be noted particularly in regard to Fig. 2 is that the maximum efficiency obtained is practically independent of the number of revolutions per minute, since all the losses in the blower vary directly as the cube of the number of revolutions per minute, as does also the total work done per pound of air. Consequently, the losses bear the same relation to the total input at a small number of revolutions or at a low pressure discharged against as they do for a higher number

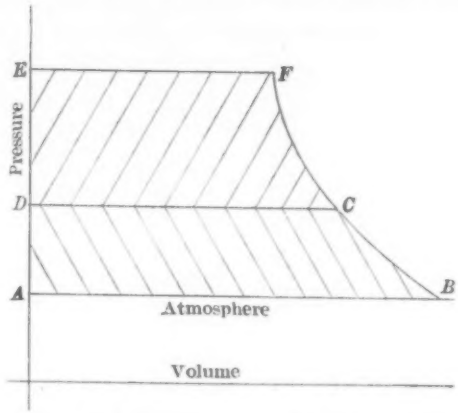

FIG. 3 DIAGRAM OF WORK DONE BY TWO IMPELLERS IN SERIES

of revolutions and higher pressure. For this reason the efficiency curve characteristic remains the same for changes of speed, but the point of maximum efficiency shifts to a smaller volume with decrease in the number of revolutions per minute.

14 As the work done per pound of air is independent of everything except the tip speed and the volume of the air entering the impeller, it is evident that the air discharged from one impeller may be supplied to a second impeller, which will add the same number of foot-pounds of work per pound of air to it, thus increasing the pressure by adding that created by the first impeller to that created by the second. The effect of this will be seen by examining Fig. 3 in which the area *BADC* represents the work done by an impeller compressing from atmospheric pressure to a pressure *D*. Where the air is discharged into a second impeller doing the same amount of work or the area as represented by the area *CDEF*, which is equal to the area *BADC*, it should be noted that, owing to the decrease of volume with increase of pressure, the actual difference of pressure created in the second impeller is greater than that in the first, so

that high pressures can be attained with a comparatively small number of impellers or stages in series. The efficiency of such a combination of impellers placed in series is exactly the same as though each impeller were an independent unit, the total efficiency being the same as that of the individual stages.

15 Figs. 4, 5, 6 and 7 show diagrammatically the construction

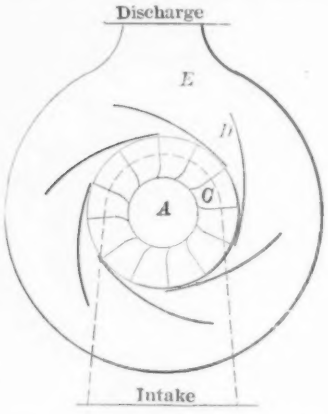

FIG. 4 GUIDE VANE BLOWER

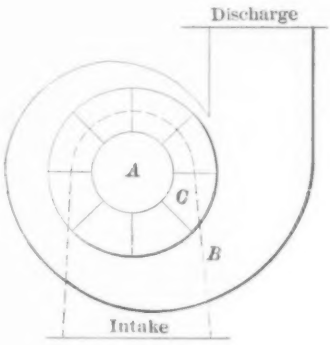

FIG. 5 VOLUTE BLOWER

and characteristics respectively of the guide vane and volute types of blower. As will be noted by examining Fig. 4, the guide vanes consist of a number of curved vanes surrounding the impeller or rotor and pointing in the direction of rotation. Since the air discharged from the impeller enters the diffusion vanes at the narrowest portion at a high velocity and leaves at the point of greatest cross-section with a greatly reduced velocity, the difference in kinetic energy between the point of entrance and exit is partly utilized in increasing the pressure of the air and partly in eddies and friction, the latter part reappearing in the form of heat. The characteristic of this type of blower is seen in Fig. 6, which shows that the pressure at first increases as the volume of discharge increases until it reaches a maximum, after which the pressure again falls off, and it will also be noted that at constant discharge (and constant revolutions per minute) the pressure does not remain constant but is continually fluctuating between limits, as shown by the shaded lines, which usually represent about 10 to 15 per cent of the total pressure created. The reason for this will be understood by examining Fig. 8.

16 For the sake of illustration, instead of assuming the air to

leave the impeller, imagine it to issue from a nozzle through which the air is supplied at a pressure P_1 and expands to a pressure P_2 such that the velocity at the mouth of the nozzle is exactly equal to the velocity with which it leaves the tips of the impeller blades and enters the guide vanes. Further, in place of the guide vanes, assume a straight diffusion tube discharging into a receiver R. The pressure in it is controlled by means of a valve or other device. Hence, from the assumptions made, the diffusion tube when supplied with air as stated is an exact equivalent of the guide vanes in the blower shown

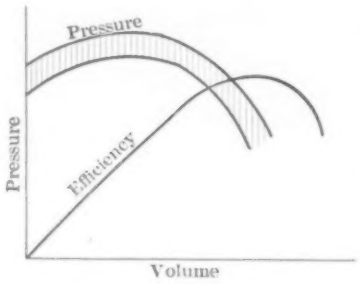

FIG. 6 GUIDE VANE BLOWER CHARACTERISTICS

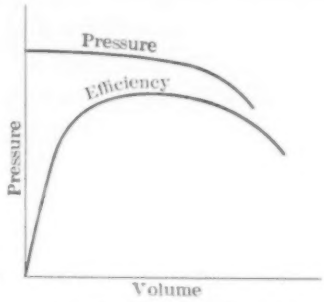

FIG. 7 VOLUTE BLOWER CHARACTERISTICS

in Fig. 4. Now, since the area of the diffusion tube increases, the velocity of the air must decrease, with the result, as previously mentioned, that the kinetic energy is partly converted into pressure, as indicated by the diagram beneath the diffusion tube. The line A_1 is the final pressure resulting from the velocity attained by the air in expanding in nozzle from pressure P_1 at A to the back pressure P_2, and the line B_2 represents the final pressure of the air when expanded from the initial pressure B to the final pressure P_2. Now it will be evident from the method in which this velocity conversion takes place that there can be but one pressure of equilibrium for any given velocity with which the air enters the diffuser. Furthermore, if the pressure P_3 in the receiver R is built up to the point where air can no longer enter the inlet of the diffuser, the pressure in the receiver must break down, for with no air entering the diffuser the velocity becomes zero, and velocity conversion is no longer possible. The result of this condition is that in a centrifugal blower fitted with guide vanes, under certain conditions the pressure may break down to the point where the only pressure available at the discharge is that due to the impeller alone, and even with normal rates of discharge there is also a tendency for the diffuser vanes to build up a

pressure higher than that in the receiver R, resulting in a constant building up and breaking down of the pressure, as shown in Fig. 9.

17 The rapidity with which these variations occur depends upon the volume of the receiver space and pipe line into which the blower is discharging. The smaller the receiver volume the more rapid the fluctuations and the smaller their extent. With very large receiver volumes and especially with a number of such blowers operating in parallel, fluctuations may occur in which the pressure breaks down to that of the impeller alone. To obviate this, it is customary to

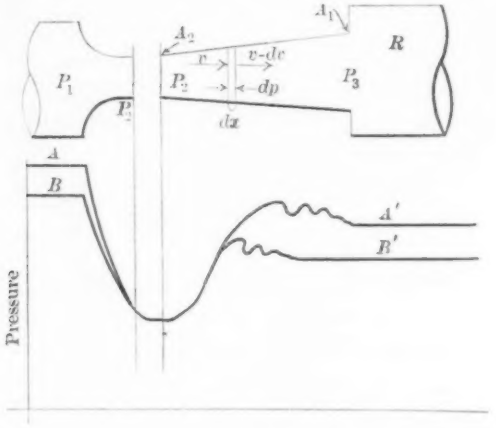

FIG. 8 PRESSURE VARIATIONS, NOZZLE AND DIFFUSION TUBE

throttle such blowers at the intake, so as always to operate them below the lower pressure line in Fig. 6, thus avoiding the fluctuations of pressure.

18 The fact that no pressure is created by the diffuser tube when no air is being delivered is the reason for the increase of pressure with increase of volume delivered up to a certain point beyond which the diffusion tubes are no longer of sufficient exit areas to reduce the final velocity to the necessary point, which in addition to the increased friction at the higher velocities causes a dropping off of the pressure line, as shown in Fig. 6. The result of this is that the efficiency curve rises very slowly at first and becomes a maximum over a comparatively very short range, when it immediately begins to drop off very rapidly.

19 The above actions will be better understood by examining the theory of diffusion vanes, in connection with Fig. 8, as follows:

20 Let the area of the smaller or entrance end of the diffuser be A_2 and the exit area be A_1; also assume that no heat is received by or given up by the diffusion tube, in which case the compression will be adiabatic (though in practice not necessarily isentropic) and the relation between pressure and volume at any point will be determined by the equation

$$pv^\gamma = p v_2^\gamma \quad \text{or} \quad v = \left(\frac{p_2}{p}\right)^{\frac{1}{\gamma}} v_2$$

21 In Fig. 8 consider an infinitely thin section of the tube dx and assume that the pressure at the point the section is taken is p, and the velocity on one side of the section is V and that the velocity on the other side is $V - dV$. Also assume that the mass of air within the section considered meets with such a resistance as will be produced by the force $dF = A\,dp$, A being the area of the tube at the section taken.

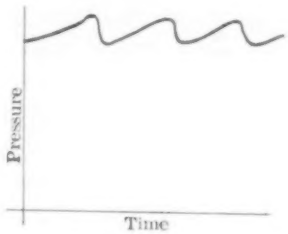

FIG. 9 PRESSURE VARIATIONS, GUIDE VANE BLOWER

22 Now the mass of the air in the section will obviously be $dw = PA\,dx$. But force is equal to mass times acceleration, hence,

$$dF = \frac{\rho A\,dx}{g}\,\frac{v - dv}{dt} = \frac{\rho A\,dx}{g}\,\frac{d^2x}{dt^2}$$

and also

$$dF = A\,dp$$

Hence equating these two expressions for dF

$$A\,dp = \frac{\rho A\,dx}{g}\,\frac{d^2x}{dt^2}$$

but $\rho = \dfrac{1}{v}$

hence

$$A\,dp = \frac{A\,dx}{v_2 \left(\dfrac{p_2}{p}\right)^{\frac{1}{\gamma}} g}\,\frac{d^2x}{dt^2}$$

or

$$g v_2 \; p_2^{\frac{1}{\gamma}} \frac{dp}{p^{\frac{1}{\gamma}}} = \frac{d^2x}{dt^2} \; dx$$

which may be integrated between V_1 and V_2 and p_2 and p_1, and will give

$$2g \; \frac{\gamma}{\gamma - 1} \; RT_2 \left[\left(\frac{p_1}{p_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] = - \left[V_1^2 - V_2^2 \right] \quad \ldots\ldots\ldots [3]$$

that is, since kinetic energy has disappeared into potential energy the sign is negative. The most interesting part of this equation is that an examination shows that the kinetic energy lost in changing from a velocity V_1 to velocity V_2 is exactly the same as that given up in adiabatic expansion from the pressure P_1 to the pressure P_2. In other words, neglecting friction eddies and shock, for some fixed relation between the velocity of approach and final pressure, as indicated by the formula, the theoretical velocity conversion efficiency of a diffusion tube is unity, and constant delivery is likewise theoretically possible.

23 Returning now to the volute blower shown in Fig. 5, which is not fitted with guide vanes, an entirely different characteristic is obtained, the theoretical characteristic of this blower being a perfectly straight line, that is, constant pressure for any delivery, neglecting, of course, the energy put into the air in order to move it through the blower. In this type of blower, as there is no velocity conversion in the volute, the pressure remains the same at all rates of delivery except for the increased frictional resistance with higher velocities and larger discharge rates, which results in a gradual decrease of the pressure as shown in Fig. 7.

24 However, the most remarkable point in this design of blower is the rapidity with which the efficiency curve rises with increase of discharge rate and the wide range over which it remains practically constant. Though not plotted in scale, the pressure characteristics and efficiency characteristics shown in Fig. 7 are relatively exact, as found in the two types of blower, the maximum efficiency obtainable in both the volute and guide vane being for both purposes identical, provided, of course, that both are properly designed; but the point which should be noted particularly is that in commercial operation the value of a blower operating on fluctuating load depends not upon the maximum efficiency obtained, but upon the area under the efficiency curve, since this is a true measure of the average efficiency over a range of load.

25 The results from two blowers on actual test are given in Fig. 10, which clearly show the difference in the characteristics and the area under the two efficiency curves.

26 In order to find the pressure created in a free vortex, assume a tube extending from R_1 to R_3, Fig. 11, of area A and take a lamina

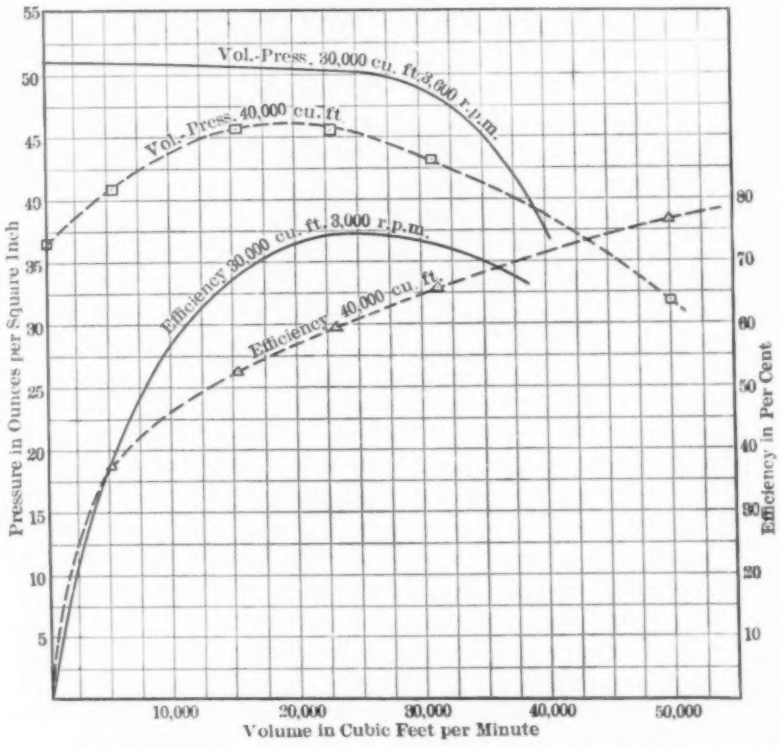

FIG. 10 CHARACTERISTICS OF TWO BLOWERS UNDER DIFFERENT EFFICIENCY CURVES

of this tube of thickness dr at any radius r. Then the centrifugal force acting on this small mass of air will be

$$dF = Adp = \frac{\rho A dr \mu^2}{gr} \quad \text{or} \quad vdp = \frac{dr\mu^2}{gr}$$

But μ is inversely proportional to the radius, hence the velocity μ at any point will be $\frac{\mu_1 R_1}{r}$ where μ_1 is the tip velocity of impeller. Substituting the value of μ,

$$vdp = (\mu_1 R_1)^2 \frac{dr}{gr^3}$$

and integrating between the limits p_2 and p_3 and $r = R_1$, and $r = \infty$,

$$\frac{\gamma}{\gamma - 1} 2g\, p_2 v_2 \left[\left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] = -\frac{\mu_1^2 R_1^2}{\infty^2} + \frac{\mu_1^2 R_1^2}{R_1^2} = \mu_1^2$$

which shows that for a 100 per cent conversion efficiency the volute must be of infinite diameter. For any given radius of volute R_3 the work done by the velocity conversion in foot-pounds will be

$$W = \frac{\mu_1^2}{2g} \left[1 - \left(\frac{R_1}{R_3} \right)^2 \right]$$

or the velocity conversion efficiency is

$$E = 1 - \left(\frac{R_1}{R_3} \right)^2$$

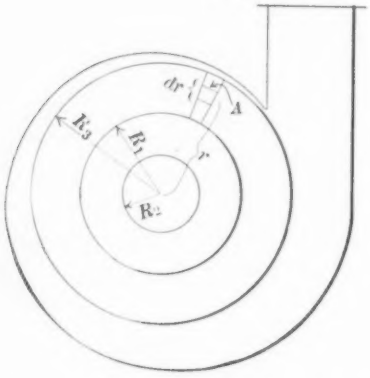

FIG. 11 PRESSURE DIAGRAM IN A VOLUTE BLOWER WITH A FREE VORTEX

27 Now the reason for the statement that the pressure remains practically constant in a blower having a free vortex will be evident from the preceding mathematical demonstration, since it follows that as the pressure created in the vortex is due to the centrifugal force acting on the fluid in the vortex, neglecting friction and eddies, the pressure created must be independent of the volume delivered. This must be true for the further reason that the volume delivered does not enter into the mathematical analysis as a function of the work done, whereas, in the mathematical deduction of the equation of flow (or work done) in a diffusion tube, the volume enters the equation in the expression $(V_1^2 - V_2^2)$.

28 As previously stated, neglecting friction, the energy transformation taking place in a free vortex remains constant regardless of the volume delivered, even though the delivery is reduced to zero,

and likewise the efficiency is for the same reasons constant, independent of the volume delivered. This explains the much higher efficiency of the free vortex blower at partial rating than is obtained in a blower fitted with guide vanes. Further, when the element of friction is considered, the free vortex blower has a considerable advantage over the diffusion vane blower because the surface exposed to friction is very much smaller.

29 The preceding remarks are not based on theory alone, since the writer has made a large number of tests with pitot tubes and thus verified the above statements. These tests showed that, when properly constructed, theory and practice agree to a surprising extent.

30 In a discussion before the Society some time ago the design of the impellers used by the Westinghouse Machine Company was criticized as inefficient because the blades are radial and have no curvature at their inlet ends for picking up the incoming air without shock losses.

31 It has been shown that if the blades of an impeller extend all the way into the center of the shaft, the work done by compression in the rotor is just one-half of the total work done on the air, the other half being represented by the kinetic energy in the air leaving the impeller with the velocity of the blade tips, that is, energy represented by $\frac{\mu_1^2}{2g}$ per lb.

32 In actual practice the inner diameter of the blades, or "eyes," is usually about one-half the diameter of the impeller, or say just one-half, then the velocity of the entrance edges of the blades is one-half the tip velocity, that is, $\mu_2 = \frac{1}{2}\mu_1$. But the total energy put into the air is $2\left(\frac{\mu_1^2}{2g}\right)$ per lb., and since the velocity at the blade entrance is $\frac{1}{2}\mu_1$ the maximum loss from shock, if the air hits the blades at right angles, as in Fig. 12, would be $\frac{(\frac{1}{2}\mu_1)^2}{2g}$ and the percentage loss of the total energy put into the blower would consequently be

$$100\frac{\frac{(\frac{1}{2}\mu_1)^2}{2g}}{2\left(\frac{\mu_1^2}{2g}\right)} = 12.5$$

33 The preceding is on the assumption that all the velocity of

the air relative to the blade is destroyed, which, however, is far from the truth.

34 Since it is necessary for the air to have a radial velocity in order to pass through the impeller (usually the radial velocity of the air is about one-quarter of the tip velocity, or one-half of the velocity of the entrance of the blades), instead of striking the blades at right angles, it enters the blades about as shown in Fig. 13 at an angle of 45 deg. Hence, if a total loss by shock of the velocity component perpendicular to the blade is still assumed, the loss at the entrance is reduced from 0.125 of the total energy to about 0.06 of the total energy put into it. However, it is known that when air strikes a surface at an angle, there is a tendency to build up a bank of air, as shown in Fig. 14, and since the flow is continuous, all the energy is not lost by shock. Hence, it is very conservative to estimate the

FIG. 12 STREAM LINES, AIR HITTING BLADES AT RIGHT ANGLES

FIG. 13 STREAM LINES, AIR HITTING BLADES AT 45 DEG.

FIG. 14 BUILDING UP AN AIR-BANK, AIR HITTING BLADES AT AN ANGLE

loss at the entrance with radial blades as not exceeding 1 or 2 per cent. This is also verified by the tests which have been made, as it was impossible to find the loss at the entrance because of its being so small compared with the total energy put into the air.

35 Another reason why the loss is practically nothing in the Westinghouse design is that the inlet of the blowers is made in the form of a volute and forced vortex as shown in Fig. 15, thus causing the air to rotate in the same direction as the impeller before entering the impeller, and the proportions of this volute are so made that at normal load the air enters the impeller blades tangentially without any shock loss whatever.

36 Since centrifugal blowers for all except the very lowest pressures must run at comparatively high tip speeds, it becomes imperative that the rotor construction shall be of the very best design

possible, and all the stresses in the blades must be radial. Hence, a straight radial blade is the only one which can be used except at the very lowest speeds.

37 No riveted or built up construction can be satisfactory and only an impeller of the double-flow type milled out of the solid should be tolerated. Milling the rotor in halves and riveting them together is also a very poor construction, and has only cheapness to recommend it. The disk should be of such section as to give uniform stress throughout, and there should be no hub or hole for the shaft.

THE TESTING AND CALCULATION OF CENTRIFUGAL BLOWERS

38 The following notes give the readings to be taken, together with the formula used for working out the results of tests, as well as

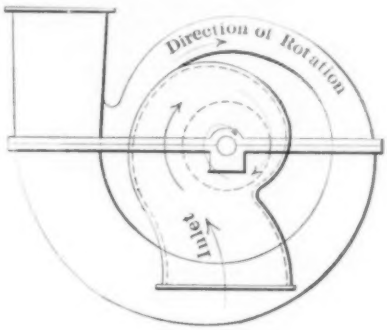

FIG. 15 INLET OF BLOWER, WESTINGHOUSE DESIGN

the method of correcting to standard or any selected contract conditions.

39 The data which must be obtained for a complete test of a centrifugal blower comprise

a brake horsepower input to the blower shaft
b inlet temperature
c inlet pressure of the air entering the blower
d barometer reading
e static pressure in the blower discharge, and temperature in the blower discharge
f total head blown against, i. e., static head plus velocity by means of a pitot tube placed above the discharge nozzle, and flush with its outlet.

40 In obtaining the correct inlet temperature to the blower, the

temperature should be taken by means of a number of thermometers placed in different portions of the intake duct, so as to obtain a true average temperature, and precautions should be taken to see that no air, other than that passing the thermometers, is entering the blower intake, such as, for instance, air drawn from the basement below the engine-room floor.

41 The pressure in the blower discharge should be taken at a point as shown in Fig. 18, in which the distance h should be at least four or five times the outlet diameter of the nozzle, in order to avoid a reduction of pressure, which occurs near the nozzle.

42 The temperature also should be taken at a corresponding point by inserting one or more thermometers into the discharge.

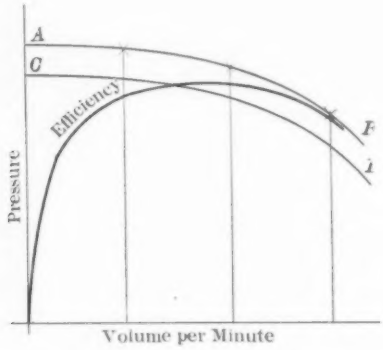

FIG. 16 PRESSURE AND EFFICIENCY CURVES, CENTRIFUGAL BLOWER

This can be most conveniently done by sticking the thermometers through rubber inserted in the discharge pipe.

43 The method of obtaining the total pressure, or static plus velocity head, will be seen from Fig. 18. Either mercury U-tubes or water manometers may be used for measuring the discharge pressure. In either case, the difference between the two sides of the U, i. e., the head in inches of mercury, should be corrected for the temperature of the U-tube.

44 The test nozzles should be made about as shown in Fig. 18; the radius or curvature of the approach to the nozzle should be equal to d, in which case the coefficient of the nozzle is for all practical purposes 100 per cent, or equal to the theoretical discharge. It has been generally agreed that a nozzle coefficient of 98.5 is, if anything,

probably 1 per cent low, and hence has been adopted where it is not convenient to calibrate the nozzle used.

METHOD OF CALCULATION

45 In the following calculation let

P_2 = absolute inlet pressure to the blower in inches of mercury (i. e., barometer reading)

P_1 = total head (static plus velocity) in inches of mercury, plus the barometer reading in inches of mercury

T_2 = absolute inlet temperature in deg. fahr.

T_1' = temperature in the discharge in deg. fahr. absolute

T_1 = theoretical temperature in the piping after adiabatic compression

T_2' = temperature at the mouth of the discharge nozzle.

46 The work done in foot-pounds per pound of air, in compressing from P_1 to P_2 will be expressed by the formula

$$W = \frac{\gamma}{\gamma - 1} RT_2 \left[\left(\frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad \ldots\ldots [4]$$

in which R has a value of 1.4065, and the work done per pound of air in expanding from the pressure P_1 to atmospheric pressure in the nozzle will be

$$W = \frac{\gamma}{\gamma - 1} RT_2' \left[\left(\frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \quad \ldots\ldots [5]$$

in which

$$T_2' = \frac{T_1'}{\left(\frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}}} \quad \ldots\ldots [6]$$

Hence, the velocity of the air in feet per second will be

$$V = 8.025 \sqrt{\frac{\gamma}{\gamma - 1} RT_2' \left[\left(\frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]} \quad \ldots\ldots [7]$$

Accordingly, the volume of discharge in cubic feet per second will be

$$\text{Volume per second} = AV \quad \ldots\ldots [8]$$

in which A is the area of the mouth of the nozzle in square feet times the nozzle coefficient, which, as above stated, for nozzles properly made, is practically unity, or say 99 per cent. The total work done in foot-pounds per minute will be equation [4] $\times$ equation [8] $\times$ 60 $\div$ specific volume of 1 lb. of air at the barometric pressure and temperature T_2, which may be expressed by the equation

$$W_{min}=\frac{\frac{\gamma}{\gamma-1}RT_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]\times 8.025\sqrt{\frac{\gamma}{\gamma-1}RT_2'\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]}\times A\times 60}{\frac{RT_2'}{P_2'}} \quad [9]$$

in which P_2' is the barometric pressure expressed in pounds per square foot absolute. Hence from equation [9] the theoretical air horsepower required to compress the column handled from the pressure P_2 to the pressure P_1 will be equation [9] divided by 33,000, or air h.p.

$$=\frac{\frac{\gamma}{\gamma-1}RT_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]\times 8.025\sqrt{\frac{\gamma}{\gamma-1}RT_2'\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]}\times A\times 60}{\frac{RT_2'}{P_2'}\times 33{,}000} \quad \ldots [10]$$

47 The efficiency of the blower is equal to the theoretical horsepower required to compress the given volume of air handled divided by the brake horsepower input at the shaft; hence, the efficiency of the blower is expressed by the equation

$$E=\frac{\text{air horsepower}}{\text{motor input}\times\text{motor efficiency}} \quad \ldots [11]$$

and the total overall combined efficiency will be

$$E_o=\frac{\text{air horsepower}}{\text{motor input}} \quad \ldots [12]$$

in each of which the motor input has been reduced to horsepower.

48 The blower efficiency may also be found by the temperature-pressure method in any blower which is not water-jacketed, the procedure in this case being to calculate the theoretical temperature rise from the formula

$$T_1-T_2=T_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right] \quad \ldots [13]$$

dividing it by the actual temperature rise found by subtracting the inlet temperature from the temperature in the blower discharge, which gives rise to the equation

$$E'=\frac{T_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]}{T_1'-T_2} \quad \ldots [14]$$

49 Following is the derivation and use of formula [14].

To Prove: the efficiency of a blower is measured by the formula

$$E' = \frac{T_1 - T_2}{T_1' - T_2}$$

where

T_1 = theoretical final temperature of the air calculated from the adiabatic relation between pressure and temperature

T_2 = initial temperature of entering air

T_1' = final observed temperature in actual compressor.

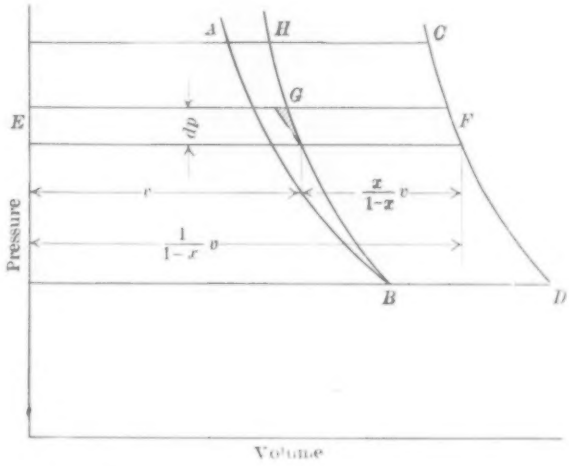

FIG. 17 WORK DIAGRAM, CENTRIFUGAL BLOWER

ANALYSIS

50 Any losses of energy which occur in a blower must reappear in some other form, namely, as heat. Hence, the various losses of whatever nature occurring in a blower may be considered equal to x. The losses include friction, eddies, leaking, etc. Referring to Fig. 17, let AB be the theoretical adiabatic compression curve and HB the actual compression line. Let the section EG be an infinitely small lamina of height dp. Since the work of vdp must be $\frac{1}{1-x}vdp$ extending the area EG to a point F on an imaginary compression curve CD. Note that vdp is the work done on the air as shown by the area of the actual compression curve and not under the adiabatic. Since, from the fundamental assumption that energy lost is converted into heat, the first law of thermodynamics

$$dQ=\frac{1}{\gamma-1}vdp+\frac{\gamma}{\gamma-1}pdv \quad \ldots\ldots\ldots\ldots\ldots\ldots [15]$$

may be used. Substituting for dQ its value, the area

$$GF=\frac{x}{1-x}vdp$$

equation [15] becomes

$$\frac{x}{1-x}vdp=\frac{1}{\gamma-1}vdp+\frac{\gamma}{\gamma-1}pdv \quad \ldots\ldots\ldots\ldots\ldots\ldots [16]$$

Integrating equation [16] between p_2 and p_1 and v_2 and v_1

$$v_1=v_2\left(\frac{P_2}{P_1}\right)^{\frac{1+\frac{x}{1-x}-\frac{x}{1-x}\gamma}{\gamma}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [17]$$

$$\beta=\frac{1+\frac{x}{1-x}-\frac{x}{1-x}\gamma}{\gamma} \quad \ldots\ldots\ldots\ldots\ldots\ldots [18]$$

Substituting the value of v found from equation [17] in the equation $dW=vdp$ and integrating,

$$W=\frac{RT_2}{(1-x)\ (1-\beta)}\left[\left(\frac{P_1}{P_2}\right)^{(1-\beta)}-1\right] \quad \ldots\ldots\ldots\ldots\ldots\ldots [19]$$

51 Equation [19] gives the work done in foot-pounds per pound of air compressed from P_2 to P_1 including all losses of whatever nature occurring in the blower. It will therefore be evident that the efficiency of the blower will be equal to the theoretical work required to compress one pound of air from P_2 to P_1 divided by the actual work required for the same compression. Hence

$$E'=\frac{RT_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]^{\frac{\gamma}{\gamma-1}}}{RT_2\left[\left(\frac{P_1}{P_2}\right)^{1-\beta}-1\right]\frac{1}{(1-x)\ (1-\beta)}} \quad \ldots\ldots\ldots\ldots [20]$$

but

$$\frac{1}{(1-x)\ (1-\beta)}=\frac{\gamma}{\gamma-1}$$

also

$$T'_1-T_2=T_2\left[\left(\frac{P_1}{P_2}\right)^{1-\beta}-1\right]$$

hence the expression for the efficiency reduces to

$$E' = \frac{\frac{\gamma}{\gamma-1} RT_2 \left[\left(\frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]}{\frac{\gamma}{\gamma-1} RT_2 \left[\left(\frac{P_1}{P_2} \right)^{1-\beta} - 1 \right]} = \frac{T_1 - T_2}{T'_1 - T_2} \quad \ldots\ldots\ldots\ldots [21]$$

52 At this point it will be well to take up the old formulae used in a number of textbooks dealing with the measurement of air, viz.,

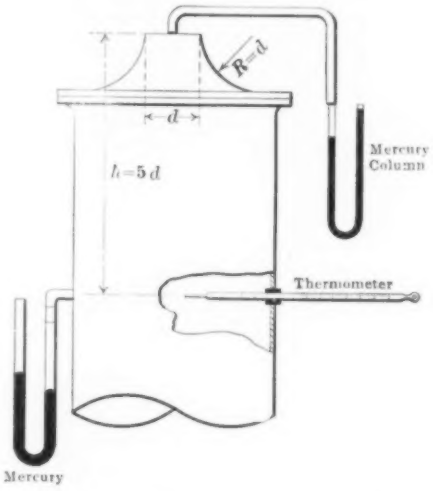

FIG. 18 TEST ARRANGEMENT FOR OBTAINING TOTAL PRESSURE

$$V = \sqrt{2gh} = 8.025 \sqrt{h} \quad \ldots\ldots\ldots\ldots\ldots\ldots [22]$$

in which h is the head through which the medium flows. In the case of air, or any gaseous medium, h being taken as

$$h = \frac{P_1 D \times 13.596}{12\rho} \quad \ldots\ldots\ldots\ldots\ldots\ldots [23]$$

in which

P_1 = static, plus velocity head, in inches of mercury
D = weight of 1 cu. ft. of water at T'_2
ρ = weight of 1 cu. ft. of air at temperature T'_2 and barometric pressure.

53 Equation [23] is substantially correct for the measurement of velocity for comparatively low pressures, but gives a velocity slightly too high when employed for pressures of 2 or 3 lb. or more, in which case the velocity, as previously pointed out, should be

calculated from the formula [7] the error arising from the use of for mula [22] being shown in Fig. 19. The work done is represented by $2gh$, being the area $BCDE$, whereas the actual area is that under the adiabatic curve represented by the curve $BCDA$.

54 The method of correcting for temperature at inlet to blower, and for changes of barometer is as follows:

55 The fundamental characteristic of the centrifugal blower is

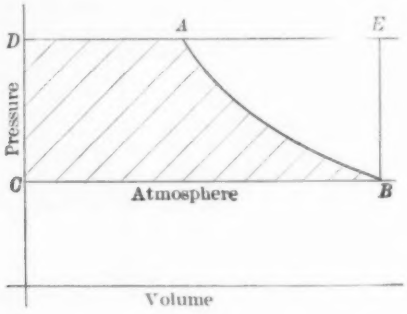

FIG. 19 WORK DIAGRAM, SHOWING DIFFERENCE BETWEEN TEXTBOOK AND CORRECTED FORMULAE

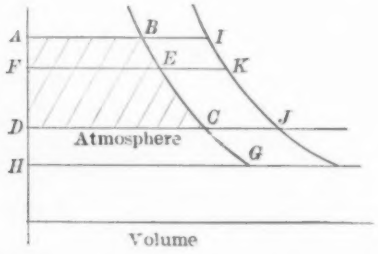

FIG. 20 INDICATOR DIAGRAMS OF CENTRIFUGAL BLOWER FOR DIFFERENT CONDITIONS OF INLET TEMPERATURE AND PRESSURE

that for any given actual volume entering the blower inlet, the blower at a given speed will take a certain number of foot-pounds of work per pound of air, regardless of the intake temperature or pressure, and the result of this is that the ordinary methods of correcting for temperatures and barometer as applied to blowing engines, does not apply to centrifugal blowers. This will be seen from Fig. 20, which shows an indicator diagram of the centrifugal blower for different conditions of inlet temperature and pressure. Let the area $CDAB$ be the compression over the standard temperature and

barometer, to some given pressure. This area will represent a fixed number of foot-pounds of work per pound of air. It is evident, therefore, that if the barometer readings were the same but the intake temperature higher, i. e., so that the initial volume were to point *J* instead of *C*, it is evident that the pressure to which the blower could compress would no longer be *A*, but some pressure *F*, such that the area *JDFK* would equal the area *CDAB*.

56 Consequently, the correction for change of temperature from *C* to *J* would not be applied as affecting the actual volume entering the blower, but as a correction to the pressure. In the same way, supposing that the barometer were low and the atmospheric line represented by *HG* at standard temperature, the area *GHFE* must again equal the area *CDAB*, and consequently the pressure would

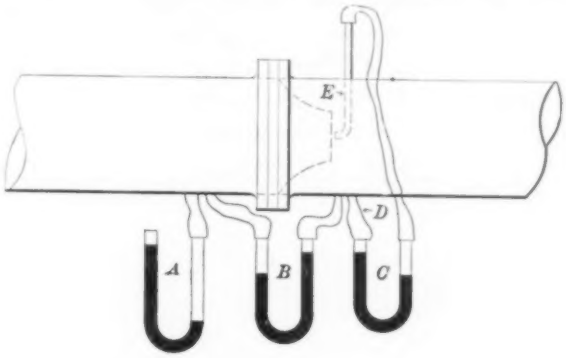

FIG. 21 ARRANGEMENT OF TEST FOR DETERMINING FLOW IN PIPE

be lower than it would be with the standard barometer. This is the correction for both inlet temperature and barometer, to the pressure of compression and not to the inlet volume. As previously stated, the work done in foot-pounds per pound of air remains constant; hence, having found the work in foot-pounds per pound of air over the actual test conditions, it is possible to substitute in equation [4], the value of the standard inlet temperature in deg. fahr. absolute, and a standard barometer reading, and in this way to calculate the final pressure in the discharge for the standard conditions.

57 If P_2' and T_2' are the absolute pressure and temperature at the inlet to which the test results are to be corrected, it follows that the corrected discharge pressure P_1' will be given by the equation,

$$P_1' = P_2' \left\{ \frac{T_2}{T_2'} \left[\left(\frac{P_1}{P_2} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] + 1 \right\}^{\frac{\gamma-1}{\gamma}}$$

which is the form to be used in all cases where the discharge pressure is to be corrected to constant conditions.

58 In Fig. 16, let AB represent the curve of pressure as determined from the original test data, and assume, for instance, that the temperature reading at the intake was higher than the standard, or that the barometer was lower than the standard, test points on the curves being represented by the crosses. The method of determining the pressure for standard conditions then would be to calculate the pressures on the line CD, as shown in Fig. 16, the curve being plotted, however, on the original volume lines.

59 When it is impossible to test a blower with the discharge going to the atmosphere, as when a nozzle is used as shown in Fig. 8, the discharge may be measured by obtaining the velocity in the discharge pipe by means of a pitot tube. These measurements are not always reliable, however, as ordinarily made, since great difficulty is experienced in obtaining the true static pressure of a moving fluid, for the reason that the "pressure end" of the tube itself disturbs the stream lines, and hence inaccurate readings are probable.

60 If a pitot tube is used in the ordinary way the pressure end should consist of a thin plate about 4 in. long by about 2 in. wide, having a slit about 1/32 in. wide and 2 in. long at its center. A collecting chamber back of the slit connected to the search pipe tends to equalize the pressure and give a true static pressure reading; but great care is necessary to see that the flat plate is exactly parallel to the stream lines at the point the pressure is desired.

61 A method of determining the flow in pipes which the writer believes is far more accurate and reliable than ordinary pitot tube readings is shown in Fig. 21. A well rounded nozzle having an area of one-fourth to three-fourths that of the pipe is bolted in between two flanges.

62 On the high-pressure side of the nozzle there is a gage A and a thermometer (not shown), and a second gage B has one leg connected to the high-pressure side of the nozzle and the other to the low-pressure side, thus giving the drop of pressure through the nozzle. A pitot search tube is fitted on the low-pressure side of the nozzle as shown in the figure. This search tube has its opening pointed directly against the flow through the nozzle, and should have a graduated scale so that its position in front of the nozzle can be accurately known.

63 The advantages of this arrangement are that the difficulty of obtaining a true pressure reading is avoided, and a number of checks

may be made on the readings taken, so that should inaccuracies occur for any reason, they can be immediately detected and the trouble located.

64 From Bernouilli's theorem, the sum of the pressure head and velocity head must always be equal, hence, with the search tube E in the position shown in Fig. 21, the pressure differences shown by gages B and C must be equal. Also, if the connection D to gage C be disconnected, the pressure indicated by gage C must equal the pressure shown by gage A. Also, if gage C is connected to the pipe by D, but the pitot tube disconnected, the sum of gages B and C must equal the pressure indicated by gage A. If these various readings check, it follows that the static pressures are being accurately measured and that the pitot tube is pointing directly into the stream.

65 Having thus checked the correctness of the measurements, the jet can be searched by moving the pitot tube across the nozzle at regular intervals and readings taken, from which the velocity at any point can be calculated by the formula

$$V = \sqrt{2g \frac{\gamma}{\gamma - 1} RT_1 \left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} \right]}$$

ewher

P_1 is the absolute pressure above the nozzle
P_2 is the absolute pressure below the nozzle
T_1 is the observed temperature above the nozzle
V is the velocity in feet per second

With V known and a known area of discharge, the volume passing can be calculated.

66 By this method, if the observations are carefully made, the error should not exceed a very small fraction of 1 per cent. It may not be out of place to add that too great care cannot be taken to make sure that there are no air leaks in the connections to the gages, as even a pinhole may cause a very considerable error in the readings.

67 In the following, in order to show the method of working up a test on a centrifugal blower, the actual readings taken during a 1-hour test on a 30,000 cu. ft. blower have been used. The results have been worked out by seven-place logarithms, using the formulae as given under tests.

68 For practical purposes, however, it is rather inconvenient to work out the values of

$$\left(\frac{P_1}{P_2} \right)^{\frac{\gamma - 1}{\gamma}} - 1$$

and

$$\left\{\frac{T_2}{T_2'}\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1\right]+1\right\}^{\frac{\gamma}{\gamma-1}}$$

by logarithms, and therefore the writer has prepared a set of curves for the value of

$$\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}-1=K$$

for various values of $\frac{P_1}{P_2}$, which may also be used for the solution of the equation for the pressure correction.

APPENDIX

TEST ON WESTINGHOUSE TURBINE BLOWER

Nozzle: $12\frac{1}{4}$ in.
Static head: $P_1 = 85.6$ in. water $+ 26.66$ in. mercury $= 32.96$ in. mercury.
Barometer $P_2 = 26.66$ in. mercury
Intake temperature: $t_2 = 46.8$ deg. fahr.; $T_2 = 507.5$ deg. fahr. abs.
Discharge temperature: $t_1 = 92.6$ deg. fahr.; $T_1 = 553.3$ deg. fahr. abs.
Total input to motor $= 432.7$ kw.
input to exciter $= 4.04$ kw.
input motor only $= 428.66$ kw.
Efficiency motor $= 89.6$ per cent taken from curve including windage
Area of nozzle: 0.817 sq. ft.

$$\left\{\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\} = 0.06317$$

W = work in ft-lb. per lb. of air

$$= 184.754 \times T_2 \times \left\{\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\} = 5923$$

T'_2 is the absolute temperature at outlet of nozzle
Volume per minute in cubic feet at temperature T'_2 = Vol. T'_2

$$\text{Vol. } T_2' = 109.079 \sqrt{T_1 \left[1 - \frac{1}{\left\{\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right\} + 1}\right]} \text{ Area} \times 60 = 30{,}659 \text{ cu. ft.}$$

$$\text{Vol.}_{0^\circ} = \frac{\text{Vol. } T'_2 \times 520.7}{T_1} \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} = 30{,}670$$

V_{60F} = Vol. of 1 lb. of air at 60 deg. fahr. and barometer

$$\text{Air h.p.} = \frac{\text{Vol.}_{60^\circ}}{V_{60F}} \times \frac{W}{33{,}000} = 373.99$$

To find V_{60}: $pv = RT$ p in lb. sq. ft. = barometer

$$V_{60} = \frac{53.35 \times 520.7}{\text{Bar.}}$$

Barometer $= 26.66$ in. mercury $= 1887$ lb. sq. ft.

Efficiency derived from temperatures:

$$E' = \frac{T_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{T_1' - T_2} = 69.997 \text{ per cent}$$

$T_1 = 553.3$

$T_2 = 507.5$

Efficiency derived from motor input:

Air h.p. = 373.99 Input to motor only = 428.66 kw.

Input to motor and exciter = 432.70 kw.

$$\text{Overall} = \frac{\text{air h.p.}}{\text{total input to motor in h.p.}} = 64.479 \text{ per cent}$$

$$\text{Blower} = \frac{\text{air h.p.}}{(\text{motor input} - \text{exciter input}) \times \text{motor in h.p.}} = 72.641 \text{ per cent}$$

$$P'_1 = 26.7\left[\frac{T_2\left[\left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{520.7} + 1\right]^{\frac{\gamma}{\gamma-1}} = 32.837 \text{ in. mercury}$$

$P'_1 - P'_2 = 32.837 - 26.7 = 6.137$ in. mercury

$= 48.23$ ounces per sq. in. corrected to 26.7 in. barometer and 60 deg. fahr. at the inlet.

RULES FOR CONDUCTING PERFORMANCE TESTS

OF

POWER PLANT APPARATUS

EMBRACING

1 Boilers
2 Reciprocating Steam Engines
3 Steam Turbines
4 Complete Steam Power Plants
5 Pumping Machinery
6 Complete Pumping Plants
7 Compressors, Blowers, and Fans
8 Locomotives
9 Gas Producers
10 Gas and Oil Engines
11 Complete Gas Power Plants
12 Waterwheels

PRELIMINARY REPORT OF POWER TEST COMMITTEE,
CODES OF 1912, THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS, NEW YORK

To the Society:

In submitting the following preliminary report of the Committee on Power Tests, which was appointed May 4, 1909, with the object of revising and extending the present testing codes of the Society relating to power plant apparatus, the Committee desires a full discussion and criticism of the report by members of the Society and others who are interested in the subject.

The Committee wishes to impress upon those offering suggestions the desirability of stating explicitly in writing the additions, omissions, or changes of language to make the report conform to the views expressed. It will be a great help to the Committee if this request is literally complied with.

It should be borne in mind that the report cannot be expected to deal with all the refinements of laboratory tests or to conform to the methods practised by every individual who engages in testing work; but it is intended to set forth the correct governing principles and serve the practical purposes of the engineering public.

When the preliminary report has been fully discussed and criticised the Committee will make a final revision for publication in the Transactions.

Respectfully submitted,

COMMITTEE ON POWER TESTS

By Geo. H. Barrus

Vice-Chairman

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FIRST SECTION: GENERAL MATTERS

PART I

INSTRUCTIONS REGARDING TESTS IN GENERAL

INSTRUCTIONS REGARDING TESTS IN GENERAL

1 OBJECT

Ascertain the specific object of the test, and keep this in view not only in the work of preparation, but also during the progress of the test, and do not let it be obscured by devoting too close attention to matters of minor importance. Whatever the object of the test may be, accuracy and reliability must underlie the work from beginning to end.

If questions of fulfillment of contract are involved, there should be a clear understanding between all the parties, preferably in writing, as to the operating conditions which should obtain during the trial, and as to the methods of testing to be followed, unless these are already expressed in the contract itself.

Among the many objects of performance tests, the following may be noted:

Determination of capacity and efficiency, and how these compare with standard or guaranteed results

Comparison of different conditions or methods of operation

Determination of the cause of either inferior or superior results

Comparison of different kinds of fuel

Determination of the effect of changes of design or proportion upon capacity or efficiency, etc.

2 PREPARATIONS

(*A*) *Dimensions*

Measure the dimensions of the principal parts of the apparatus to be tested, so far as they bear on the objects in view, or determine these from correct working drawings. Notice the general features of the same, both exterior and interior, and make sketches, if needed, to show unusual points of design.

The dimensions of the heating surfaces of boilers and superheaters to be found are those of surfaces in contact with the fire or hot gases. The submerged surfaces in boilers at the mean water level should be considered as water-heating surfaces, and other surfaces which are exposed to the gases as superheating surfaces.

In the case of condensers, feedwater heaters, and the like, the outside surfaces are to be taken. In reheaters and steam jackets, the surfaces to be considered are those exposed to the steam of lower pressure.

The dimensions of engine cylinders should be taken when they are cold, and, if extreme accuracy is required, as in scientific investigations, corrections should be applied to conform to the mean working temperature. If the cylinders are much worn, the average diameter should be found. The clearance of the cylinders may be determined approximately from working drawings of the engine. For accurate work, when practicable, the clearance should be determined by the water measurement method. (See Appendix No. 1 for water determination of clearance.)

(*B*) *Examination of Plant*

Make a thorough examination of the physical condition of all parts of the plant or apparatus which concern the object in view, and record the conditions found, together with any points in the matter of operation which bear thereon.

In boilers, for example, examine for leakage of tubes and riveted or other metal joints. Note the condition of brick furnaces, grates and baffles. Examine brick walls and cleaning doors for air leaks, either by shutting the damper and observing the escaping smoke or by candle-flame test. Determine the condition of heating surfaces with reference to exterior deposits of soot and interior deposits of mud or scale.

See that the steam main is so arranged that condensed and entrained water cannot flow back into the boiler.

Ascertain the interior condition of all steam, air, gas, or water cylinders and the condition of their pistons, and of water plungers and impellers, together with the valves and valve-seats belonging thereto. Locate vacuum leaks in exhaust piping, condenser, packings, etc., using vacuum gage or candle-flame test. Examine steam, air, gas, or water piping, traps, drip valves, blow-off cocks, safety valves, relief valves, heaters, etc., and make sure that they do not leak. Determine the condition of the blading, nozzles, and valves in steam turbines, and of buckets, guides and draft-tubes in water turbines.

See Appendices Nos. 2 and 3 for general methods of leakage examination and test.

If the object of the test is to determine the highest efficiency or capacity obtainable, any physical defects, or defects of operation, tending to make the result unfavorable should first be remedied; all fouled parts being cleaned, and the whole put in first-class condition. If, on the other hand, the object is to ascertain the performance under existing conditions, no such preparation is either required or desired.

(*C*) *General Precautions against Leakage*

In steam tests make sure that there is no leakage through blow-

offs, drips, etc., or any steam or water connections of the plant or apparatus undergoing test, which would in any way affect the results. All such connections should be blanked off, or satisfactory assurance should be obtained that there is leakage neither out nor in. This is a most important matter, and no assurance should be considered satisfactory unless it is susceptible of absolute demonstration.

See Appendix No. 3 for further details.

(*D*) *Apparatus and Instruments*

Select the apparatus and instruments specified in the Code of Rules applying to the test in hand, locate and install the same, and complete the preparations for the work in view.

The arrangement and location of the testing appliances in every case must be left to the judgment and ingenuity of the engineer in charge, the details being largely dependent upon locality and surroundings. One guiding rule, however, should always be kept in view, viz., see that the apparatus and instruments are substantially reliable, and arrange them in such a way as to obtain correct data.

A summary is given below, embracing the entire list of apparatus and instruments referred to in the various Codes, with descriptions of their leading features, methods of application and use, and, where needed, methods of calibration; these particulars being supplemented in some instances by further descriptions in the Appendix.

(a) *Weighing Scales.* For determining the weight of coal, oil, water, etc., ordinary platform scales serve every purpose. Too much dependence, however, should not be placed upon their reliability without first calibrating them by the use of standard weights, and carefully examining the knife-edges, bearing plates, and ring suspensions, to see that they are all in good order.

Other scales required in connection with test work are small scales for weighing coal-samples used in drying, and laboratory scales for analysis and calorific determinations pertaining to fuels. Such scales should be sensitive to 1/1000 of the quantity weighed.

For testing locomotives and some classes of marine boilers, where room is lacking, sacks or bags are sometimes required to facilitate the handling of coal, the sacks being previously weighed at the time of filling.

(*b*) *Water Weighing and Measuring Apparatus.*

(1) Feedwater. Wherever practicable the feedwater should be weighed, especially for guarantee tests. The most satisfactory and reliable apparatus for this purpose consists of one or more tanks each placed on platform scales, these being elevated a sufficient distance above the floor to empty into a receiving tank placed below,

the latter being connected to the feed pump. Where only one weighing tank is used the receiving tank should be of larger size than the weighing tank, to afford sufficient reserve supply to the pump while the upper tank is filling. If a single weighing tank is used it should preferably be of such capacity as to require emptying not oftener than every five minutes. If two or more are used the intervals between successive emptyings should not be less than three minutes.

In tests of complete steam power plants, where it is required to measure the feedwater without unnecessary change in the working conditions, a water meter may be employed. Meter measurement may also be required in many other cases, such as locomotive and marine service. The reliability of meters should be determined by calibration in place under conditions of use.

If a large quantity of water is to be measured, a rotary, disk, or Venturi meter, or some form of orifice measurement may be employed. In the latter case the orifice should be calibrated under the conditions of use, unless its design is such that standard formulæ and constants may be applied for determining the discharge.

See Appendix No. 4 for methods of calibrating meters and Appendix No. 5 for orifice formula.

(2) Water of Condensation Under Pressure. In measuring jacket water or any supply under pressure which has a temperature exceeding 212 deg. fahr, the water should first be cooled, as may be done by discharging it into a tank of cold water previously weighed, or by passing it through a coil of pipe submerged in running and colder water, preventing thereby the loss of evaporation which occurs when such hot water is discharged into the open air. If such water is untrapped the drain pipe should be provided with a gage glass and the outlet choked so as to keep the water in sight in the glass.

(3) Water Flowing in Streams and Canals. Weirs, current meters, float rods, etc., are required for measuring the water discharged by pumping engines, and that used by waterwheels. (For description see Appendix No. 5.)

(*c*) *Steam Measuring Apparatus.* Various forms of steam meters may be employed for measuring steam, provided such meters are properly calibrated under conditions of use, and the pulsations of pressure, if any, are not serious. For measuring the steam used by the auxiliaries of a steam plant, either individually or collectively, the orifice form of steam meter may be used, consisting of an orifice in a plate inserted between the two halves of a pair of flanges in the pipe through which the steam passes, or placed in a by-pass through which the steam is diverted, with gage pipe on either side for determining the fall of pressure. The quantity of steam represented by the various differences of pressure which occur, may be found by arranging the apparatus so as to draw steam through the orifice, and discharge it into a tank of water resting on platform scales, by which its actual weight in a given time is determined.

A plate ⅛ in. thick containing an orifice 1 in. diameter, with square edges, will discharge the approximate quantities of dry steam per hour given in Table 1, with various pressure drops, the pressure below the orifice being 100 lb. by gage.

TABLE 1 DISCHARGE THROUGH ORIFICE 1 IN. DIAMETER, 100 LB. PRESSURE

Pressure Drop, Lb. Per Sq. In.	Lb. of Dry Steam Per Hour.
½	430
1	615
2	930
3	1200
4	1400
5	1560
10	2180
15	2640
20	3050

The water-glass method affords an approximate means of determining the steam consumption of auxiliaries, and for measuring the leakages of steam and water from the boiler and its connections. (See Appendix No. 3 for description of water glass method.)

(*d*) *Gas and Air Measuring Apparatus.* Apparatus to be used for measuring gas and air embraces gas meters, gasometers, Venturi meters, orifices and pitot tubes. The standard of reference for calibrating apparatus for this purpose is the gasometer. If a pitot tube is used, and this instrument cannot be calibrated by actual measurement, the constants employed should be those obtained from a similar instrument which has been calibrated by actual reference to gasometer measurement under as nearly as possible the same conditions.

Directions for using the pitot tube for gas and air measurement are given in Appendix No 6.

In this connection attention is directed to the electric method of gas measurement proposed by Mr. C. C. Thomas and described in his paper in Volume No. 31 of the Transactions,[1] also to orifice measurements described by R. J. Durley, in Volume 27.[2]

(*e*) *Anemometers.* For determining the approximate quantity of air discharged under light pressures by fans or blowers, or the amount of air used in the combustion of fuels, an anemometer of the fan-wheel type may be used, provided all the air can be passed through one or more openings of suitable size in which the anemometer may be placed. These openings should preferably be of such size (say at

[1] C. C. Thomas, An Electric Gas Meter. Trans. Am. Soc. M. E., vol. 31, p. 655.

[2] R. J. Durley, On the Measurement of Air Flowing into the Atmosphere through Circular Orifices in Thin Plates and under Small Differences of Pressure. Trans. Am. Soc. M. E., vol. 27, p. 193.

least 15 in. sq.) that the resistance interposed by the instrument (the fan being about 2½ in. in diameter) may be neglected. The instrument, when in use, can best be supported on the end of a small rod and moved slowly across the open end, also up and down, so as to obtain an average for the whole area. In the case of very large openings the area should be subdivided into a number of equal parts, the velocity determined for each subdivision, and the results averaged.

The anemometer is usually calibrated by mounting it on the end of a long and light rod which can be revolved around a central point and its reading compared with the velocity of the rod when moved in still air. Where suitable means can be provided for calibrating the anemometer under approximately its working conditions, such calibration is much to be preferred.

The quantity of air supplied for combustion in boiler and other furnaces may be determined from the analysis of the products of combustion using the formula given in the Boiler Code, p. 1724.

(*f*) *Screens for Sizing Coal.* The dimensions of screen openings to be used for sizing anthracite coals are given in Table 2; the sizes in each case being the opening through which the specified grade will pass, and that over which it will be carried without passing through. The openings referred to are circular.

TABLE 2 ANTHRACITE COAL SIZES

Name	Diameter of opening through or over which coal will pass, in.		Name	Diameter of opening through or over which coal will pass, in.	
	Through	Over		Through	Over
Broken	4 1/2	3 1/4	No. 1 Buckwheat	9/16	5/16
Egg	3 1/4	2 5/16	No. 2 Buckwheat	5/16	3/16
Stove	2 5/16	1 5/8	No. 3 Buckwheat	3/16	3/32
Chestnut	1 5/8	7/8	Culm	3/32	..
Pea	7/8	9/16			

The sizes and grades of bituminous and semi-bituminous coals vary so much according to kind and locality that there are no standards of size for these coals which are generally recognized. (For bituminous coal sizes see Appendix No. 7.)

(*g*) *Pressure Gages.* For determining steam pressure the steam gages belonging to the plant may be used, provided they are compared with a standardized gage of the spring or mercury type and verified, due allowance being made for the head of water standing in the connecting pipe.

For determining pressure below the atmosphere, a U-tube mer-

cury gage may be employed or a spring gage may be used, provided it is referred to a standard and corrected for water in the connecting pipe. In cases where extremely high vacuums are to be measured, as in turbine practice, the absolute-pressure gage is useful, provided the exhaustion is complete and afterwards no air is admitted.

For determining steam pressure on the two sides of an orifice, two gages should be used which are carefully graduated to single pounds, or, what is better, one gage should be used and this piped up so as to connect at will to either side of the orifice. A differential gage may also be employed, indicating at once the pressure drop.

For determining the water pressure in the force main of a pumping engine, the gage should be one which is sensitive to changes amounting to ½ per cent. of the pressure indicated. If such a gage is not a part of the equipment of the plant, a special test gage should be attached.

> Between the gage and the force main a small reservoir having an air chamber should be interposed, in the manner shown in Appendix No. 8, so as to prevent undue fluctuations of the gage and allow the gage cock to be run wide open. By means of a gage glass on the side of the chamber and an air valve, the average water level may be adjusted to the height of the center of the gage, and correction for this element of variation avoided. If not thus adjusted, the reading is to be referred to the level shown, whatever this may be.

For calibrating gages indicating pressures above the atmosphere, the dead-weight testing apparatus which is manufactured by many of the prominent gage makers may be employed as a standard of comparison. It consists of a vertical plunger nicely fitted to a cylinder containing oil or glycerine, through the medium of which the pressure is transmitted to the gage. The plunger is surmounted by a circular stand on which weights, may be placed, and by means of which any desired pressure can be secured. The total weight, in pounds, on the plunger at any time, divided by the average area of the plunger and of the bushing which receives it, in square inches, gives the pressure in pounds per square inch.

Another standard of comparison is the mercury column. If this instrument is used, assurance must be had that it is properly graduated with reference to the ever-varying zero point; that the mercury is pure, and that the proper correction is made for any difference of temperature that exists, compared with the temperature at which the instrument was graduated.

For pressures below the atmosphere, an air pump or some other means of producing a vacuum is required, and reference must be made to a mercury gage. Such a gage may be a U-tube having a length of 30 in. or so, with both arms properly filled with pure mercury.

> The practice of choking the gage cock to reduce fluctuations of the gage pointed, if carried too far, is objectionable, especially when there is leakage around the plug. The pressure indicated under these circumstances is less than the true pressure. Before reading a gage of

the spring type, the free working of the mechanism within should be assured by tapping the outside of the gage and thereby slightly moving the pointer.

(*h*) *Thermometers.* Thermometers should be of the kind having graduations marked on the glass stem. Those used for temperatures above the boiling point of mercury (say 450 deg.—500 deg.) should have nitrogen in the top of the bore. They should also have a small safety bulb at the top. Thermometers constructed in this way can be used satisfactorily up to 1000 deg. fahr.

Thermometers which are used for important data should be calibrated before and after a test, by reference to standard thermometers.

Standard thermometers are those which indicate 212 deg. fahr. in steam escaping from boiling water at the normal barometric pressure of 29.92 in. (referred to 32 deg.), the whole stem up to the 212 deg. point being surrounded by the steam; and which indicate 32 deg. fahr. in melting ice, the stem being likewise completely immersed to the 32 deg. point; and which are calibrated for points between and beyond these two reference marks. For temperatures between 212 deg. and 400 deg. fahr., the comparison of the thermometer should be made with the temperature given in Marks and Davis Steam Tables, the method required being to place it in a thermometer well surrounded by saturated steam under sufficient pressure to give the desired temperature. The pressure should be determined by a correct gage, and the thermometer should be immersed to the same extent as it is under its working condition.

A thermometer well consists of a hollow plug threaded at the upper end and screwed into a threaded hole in the top of a horizontal pipe, the lower part extending vertically into the interior of the pipe as far, if practicable, as the center. The inside diameter should be slightly larger than the outside diameter of the thermometer tube and the well should be filled with mercury or high-grade mineral oil for temperatures below 500 deg., and with soft solder for higher temperatures.

For superheated steam the immersed portion should be fluted so as to increase the area of the absorbing surface.

Thermometers are so readily broken that it is desirable in important tests to have a sufficient number on hand that in case of accident the readings will not be interrupted. These spare thermometers should preferably be calibrated.

(*i*) *Barometers.* For important or extremely accurate steam tests and for gas engine tests the pressure of the atmosphere should be taken, either by a mercurial or Aneroid barometer, and the reading from this instrument, reduced to pounds pressure per square inch, should be employed in determining the absolute steam pressure. In many cases it is sufficient to refer to the daily records of the nearest station of the Government Weather Bureau. Aneroid barometers may be readily calibrated by comparing them with a mercury barometer, making proper temperature corrections.

(*j*) *Hygrometers.* In tests where the hygrometric conditions of the atmosphere play an important part, it is necessary to use a hygrometer. The ordinary instrument consisting of a wet and dry bulb thermometer is suitable for the purpose. The hygrometric tables

of the Government which may be obtained from the Weather Bureau, U. S. Dept. of Agriculture, Washington, D. C., can be referred to in making the required computations. See Appendix No. 9.

(*k*) *Pyrometers.* Metallic pyrometers used for determining high temperatures must be handled cautiously owing to the difficulty of exposing the whole of the stem to the current of gas, the temperature of which is to be determined. Electric pyrometers either of the thermo-couple or resistance type are satisfactory for this work where the temperatures to be observed are not too high, say not over 2000 fahr. Instruments of this kind can readily be calibrated by comparing them at low ranges of temperature under proper conditions with a mercurial thermometer, (both being placed for example in a current of hot air the temperature of which is under control), and their accuracy at higher temperatures judged by plotting the results on a chart, finding the curve of error, and continuing the curve to the higher ranges desired. For extremely high temperatures such as that of a boiler or producer furnace, the optical, pneumatic, and radiation pyrometers may be used. See Appendix No. 10.

(*l*) *Draft Gages.* The simplest form of draft gage is the ordinary U-tube. When the tube is kept clean and the two legs are close together with the scale extending at least to the center of each leg, it gives satisfactory indications. For measuring small amounts of draft some form of multiplying gage may be employed. One of the simplest multiplying instruments consists of a U-tube in which one leg is inclined from the horizontal and the amount of multiplication varies inversely as the angle of inclination, the tube being filled with a light mineral oil. Various satisfactory instruments having the multiplying feature are on the market. These can readily be calibrated by comparison with the simple U-tube gage when indicating a high draft, say one inch or more. It is preferable to use alcohol instead of water in the U-tube, and make allowance for the difference of specific gravity.

(*m*) *Steam Calorimeters.* The most satisfactory instruments for determining the amount of moisture in steam are calorimeters that operate upon the throttling principle, or that combine the throttling and separating principles; an orifice being used of such size as to throttle to atmospheric pressure, with two thermometers, one showing the temperature above the orifice and the other that below it. If no commercial make of calorimeter is available on a test, an instrument of the throttling type can be made of pipe fittings as shown in Appendix No. 11.

Directions for applying sampling nozzles are given in Part III, Page 1713.

Further references to Steam Calorimeters are included in Appendix No. 11.

(*n*) *Coal Calorimeters.* To determine the total heat of combustion of a sample of coal or other fuel, the best form of calorimeter to use is

one in which the fuel is burned in an atmosphere of oxygen gas. The Mahler type of calorimeter is recognized as the most complete and accurate apparatus of this kind. Where the engineer conducting a test does not have this instrument, or some other reliable calorimeter at hand, the heat units can be determined by sending samples to a testing laboratory where such instruments are used.

For description of one form of the Mahler Calorimeter see Appendix No. 12.

(*o*) *Gas Calorimeters.* The total heat of combustion of gas should be found by burning the gas in the Junker calorimeter, described in Appendix No. 13.

(*p*) *Coal Analysis Apparatus.* The analyses commonly made are what are termed "proximate" analyses. For complete determinations of the quality of coal, it is necessary also to make the "ultimate" analysis. Approved methods of analysis are briefly described in Appendix No. 14.

(*q*) *Gas Analysis Apparatus.* For determining the composition of flue gases in ordinary boiler work one of the simplest and most convenient instruments is the Orsat apparatus. This instrument can readily be used by the person conducting a test, or by some assistant whom he directs.

For determining the hydrogen and other unburned combustible matter in the flue gases, and for general gas analysis, the Hempel apparatus, or some modification thereof, is required and the work should be entrusted to a person who is familiar with all phases of the subject. These instruments are briefly described in Appendix No. 15.

(*r*) *Appliances and Methods Pertaining to Smoke Determination.* No wholly satisfactory methods for either quantitative or qualitative smoke determinations have yet come into use, nor have any reliable methods been established for definitely fixing even the relative density of the smoke issuing from chimneys at different times. One method commonly employed which answers the purpose fairly well, is that of making frequent visual observations of the chimney at intervals of one minute or less for a period of one hour and recording the observed characteristics according to the degree of blackness and density, and giving to the various degrees of smoke an arbitrary percentage value rated in some such manner as that expressed in Table 3.

TABLE 3 SMOKE PERCENTAGES

Dense black	100
Medium black	80
Dense gray	60
Medium gray	40
Light gray	20
Very light	5
Trace	1
Clear chimney	0

The color and density of smoke depend somewhat on the character of the sky or other background, and on the air and weather conditions obtaining when the observation is made, and these should be given due consideration in making comparisons. Observations of this kind are also subject to personal errors and errors of judgment. Nevertheless, these methods are useful, especially when the results are plotted, according to the percentage scale determined on, so that a graphic representation of the changes can be shown.

Various forms of charts and clouded glass arrangements for comparing and fixing smoke densities have been proposed, and to some extent used, but these have proved more or less unsatisfactory and they are subject to personal errors, and to sky, wind, and weather conditions, the same as the simpler method above described.

Among the chart methods referred to, the use of the Ringelmann smoke chart is perhaps the most familiar. This is shown in Appendix No. 16.

Another method of smoke determination consists in the use of a narrow flat metal plate suspended in the flue, the character of the smoke being indicated by the amount and quality of the soot and dust deposited upon the plate in a given time. This method, like others, is useful in furnishing a means of comparison in different cases rather than a means of exact determination.

See Appendix No. 17 for further description.

(*s*) *Indicators.* To determine the amount of power developed by an engine, whether a steam engine or an internal-combustion engine, the instrument required is the widely-used steam engine indicator. One or more of these instruments is attached to the cylinder or cylinders of the engine, and operated from the cross-head or main shaft by the use of proper driving rig. As to the selection of the make of instrument, it should be one which is in all respects of first-class construction.

For fuller particulars see Appendix No. 18.

(*t*) *Planimeters.* To determine the area of indicator diagrams from which to ascertain the mean effective pressure, it is convenient to use some form of planimeter. The simplest and probably the most desirable instrument is the familiar Polar planimeter, in which the area is registered in square inches.

It is desirable to calibrate a planimeter from time to time by running it over a figure having a known area, such as a right angle triangle of say 4 in. in length and 2 in. in height, observing whether it checks with the computed area.

(*u*) *Tachometers and Other Speed Measuring Apparatus.* For determining the speed of revolution of an engine shaft, especially where the speed exceeds 300 r.p.m., a convenient instrument is a tachometer which continuously indicates on a dial the number of turns per minute. This instrument can be arranged to have a permanent location and to be operated continuously when the engine is running, or it can

be a portable instrument which is held in the hand and applied for the time being to the end of the shaft. These instruments are of two general classes, viz: fly-ball and electro-magnetic.

These instruments should be calibrated by comparison with the record obtained by counting with the watch and a speed recorder or indicator, the number of turns per minute.

The determination of variation of speed during a single revolution, or the effect due to sudden changes of the load, is desirable, especially in engines driving electric generators used for lighting purposes. There is no recognized standard method of making such determinations, and if they are desired the method may be devised to suit the requirements.

One method suggested for determining the instantaneous variation of speed which accompanies a change of load, is described as follows: A screen containing a narrow slot is placed on the end of a bar and vibrated by means of an electric current. A corresponding slot in a stationary screen is placed parallel and nearly touching the vibrating screen, and the two screens are placed a short distance from the flywheel of the engine in such a position that the observer can look through the two slots in the direction of the spokes of the wheel. The vibrations are adjusted so as to conform to the frequency with which the spokes of the wheel pass the slots. When this is done the observer viewing the wheel through the slots sees what appears to be a stationary flywheel. When a change in the velocity of the flywheel occurs, the wheel appears to revolve either backward or forward according to the direction of the change. By careful observations of the amount of this motion, the angular change of velocity during any given time is revealed.

(*v*) *Friction Brakes.* The power delivered by an engine may be determined by the application of a Prony brake to the rim of the flywheel. The friction device may consist of a simple band or rope, a number of ropes, or a series of blocks, encircling the wheel. Weighing scales either of the platform or spring type are provided for showing the strain. For long runs the wheel is made with interior flanges for holding water to keep the rim cool.

The most satisfactory brake for absorbing and measuring power is some form of water friction brake. The advantage of a water brake is that it can be employed equally well for large or small amounts of power, and it is necessarily kept cool by the water upon which it depends for its operation.

Several brakes are described in Appendix No. 19.

(*w*) *Dynamometers.* Dynamometers furnish means for determining the amount of power delivered by an engine under working conditions. In the case of a mill engine it is the power transmitted from the main shaft of the engine to the driving shaft of the mill. If this power is carried through a belt, the dynamometer measures the net amount of force transmitted. In a marine engine driving a screw

propeller through a long shaft, the dynamometer shows the torsional strain on the shaft at a point as near as practicable to the engine. In a locomotive the dynamometer measures the amount of pull on the draw-bar through which the power is transmitted to the first car of the train.

See Appendix No. 20 for details regarding dynamometers.

(*x*) *Electrical Instruments.* The output of an engine driving an electric generator should be determined by the use of a set of electrical instruments supplementary to the switchboard equipment, and connected to the generator, so as to be thrown in and out of circuit by means of switches. These instruments should include the shunts, transformers, or multipliers, which belong to them, and should be such as have been verified by comparison with known standards based on those of the U. S. Bureau of Standards at Washington. The instruments selected should be as nearly "dead-beat" as practicable, that is, the pointer should come to rest at once, after changes of load.

The instruments required for a direct-current generator, where the load is substantially constant, embrace a single ammeter and a single voltmeter. Where the load is rapidly fluctuating, a watt-hour meter is also needed.

Those required for a single-phase alternating-current generator carrying a substantially constant load, embrace one ammeter, one voltmeter, and one single-phase wattmeter. For a rapidly fluctuating load a watt-hour meter should also be used. For a two-, or three-phase, alternating-current generator, with substantially constant load, one ammeter and one voltmeter, each arranged for ready connection at will to any phase, and two single-phase wattmeters, are required. Where the load is rapidly fluctuating a polyphase watt-hour meter is also needed.

In the case of an alternating-current generator an additional voltmeter and ammeter should be used for measuring the current for excitation, unless the main instruments are connected so as to register the net output.

The instruments should be so located with reference to the generator, switchboard, and neighboring conductors, as to avoid the disturbing effects due to stray fields. If the switchboard equipment is complicated, the wiring should be connected under the supervision of one who is familiar with all the details of the switchboard, its connections, and the surrounding cables.

Watt-hour meters should be verified in place and under working conditions. Directions for calibrating these meters and further details regarding the application and use of electrical instruments, determination of power factor, etc., are given in Appendix No. 21.

(*y*) *Water Rheostats.* For tests of engines driving electric generators under certain fixed loads it is often necessary to regulate the output of current by the use of a water rheostat. One of the simplest

and most effective forms of this apparatus for direct-current work consists of a coil of wire wound spirally on a wooden reel immersed in a stream of running water, or in a tank of some natural body of water where circulation can be provided for, arranged with a system of switches so that more or less of this resistance can be turned on as may be required. A No. 12 B. G. iron wire 2000 ft. long, thus used, will handle a current of 1600 amperes at 250 volts.

Other rheostats are described in Appendix No. 22.

(z) *Steam Tables.* Quantities depending upon the properties of saturated steam which are used throughout the Codes, such as B.t.u. per pound of steam, temperatures corresponding to various pressures, etc., are based on Marks and Davis's tables; and those involving the specific heat of superheated steam, on Knoblauch and Jakob's experiments.

3 MISCELLANEOUS INSTRUCTIONS

The person in charge of a test should have the aid of a sufficient number of assistants, so that he may be free to give special attention to any part of the work whenever and wherever it may be required. He should make sure that the instruments and testing apparatus continually give reliable indications, and that the readings are correctly recorded. He should also keep in view, at all points, the operation of the plant or part of the plant under test and see that the operating conditions determined on are maintained and that nothing occurs, either by accident or design, to vitiate the data. This last precaution is especially needed in guarantee tests.

Before a test is undertaken, it is important that the boiler, engine, or other apparatus concerned, shall have been in operation a sufficient length of time to attain working temperatures and proper operating conditions throughout, so that the results of the test may express the true working performance.

It would, for example, be manifestly improper to start a test for determining the maximum efficiency of an externally fired boiler with brick setting, until the boiler had been at work a sufficient number of days to dry out thoroughly and heat the brick work to its working temperature; and likewise improper to begin an engine test for determining the performance under certain prearranged conditions until those conditions had become established by a suitable preliminary run.

An exception should be noted where the object of the test is to obtain the working performance, including the effect of preliminary

heating, in which case all the conditions should conform to those of regular service.

In preparation for a test to demonstrate maximum efficiency, it is desirable to run preliminary tests for the purpose of determining the most advantageous conditions.

4 OPERATING CONDITIONS

In all tests in which the object is to determine the performance under conditions of maximum efficiency, or where it is desired to ascertain the effect of predetermined conditions of operation, all such conditions which have an appreciable effect upon the efficiency should be maintained as nearly uniform during the trial as the limitations of practical work will permit. In a stationary steam plant, for example, where maximum efficiency is the object in view, there should be uniformity in such matters as steam pressure, times of firing, quantity of coal supplied at each firing, thickness of fire, and in other firing operations; also in the rate of supplying the feedwater, in the load on the engine or turbine, and in the operating conditions throughout. On the other hand, if the object of the test is to determine the performance under working conditions, no attempt at uniformity is either desired or required unless this uniformity corresponds to the regular practice, and when this is the object the usual working conditions should prevail throughout the trial.

5 RECORDS

A log of the data should be entered in notebooks or on blank sheets suitably prepared in advance. This should be done in such manner that the test may be divided into hourly periods, or if necessary, periods of less duration, and the leading data obtained for any one or more periods as desired, thereby showing the degree of uniformity obtained.

The readings of the various instruments and apparatus concerned in the test other than those showing quantities of consumption (such as fuel, water, and gas), should be taken at intervals not exceeding half an hour and entered in the log. Whenever the indications fluctuate, the intervals should be reduced according to the extent of the fluctuation. In the case of smoke observations, for example, it is often necessary to take observations every minute, or still oftener, continuing these throughout the period covering the range of variations.

Make a memorandum of every event connected with the progress of a test, however unnecessary at the time it may appear. A record should be made of the exact time of every such occurrence and the time of taking every weight and every observation. For the purpose of identification the signature of the observer and the date should be affixed to each log sheet or record.

In the simple matter of weighing coal by the barrow-load, or weighing water by the tank-full, which is required in many tests, a series of marks, or tallies, should never be trusted. The time each load is weighed or emptied should be recorded. The weighing of coal should not be delegated to unreliable assistants, and whenever practicable, one or more men should be assigned solely to this work. The same may be said with regard to the weighing of feedwater.

6 DATA REPRESENTED GRAPHICALLY

If it is desired to show the uniformity of the data at a glance the whole log of the trial should be plotted on a chart, using horizontal distances to represent times of observation, and vertical distances on suitable scales to represent various data as recorded. Such a chart showing log of a boiler test is illustrated in Appendix No. 23.

It is instructive to plot the leading data on such a chart while the test is in progress.

7 REPORT

The report of a test should present all the leading facts bearing on the design, dimensions, condition, and operation of the apparatus tested, and should include a description of any other apparatus and auxiliaries concerned, together with such sketches as may be needed for a clear understanding of all points under consideration. It should state clearly the object and character of the test, the methods followed, the conditions maintained, and the conclusions reached, closing with a tabular summary of the principal data and results.

PART II

STANDARDS RELATING TO CAPACITY, EFFICIENCY, AND ECONOMY, AND RULES FOR RESERVE CAPACITY

STANDARDS

The standard units on which to base the various measures of capacity, and the standard forms of expressing efficiency and economy to which the codes apply, are assembled in Tables 4 and 5.

TABLE 4 STANDARD UNITS OF CAPACITY

a Boilers*	One pound of water evaporated into dry steam from and at 212 deg. per hour.	
b Reciprocating Steam Engines	One indicated horsepower developed in the main cylinders	i.h.p.
	One brake horsepower delivered by the main shaft	br.h.p.
c Steam Turbines	One brake horsepower delivered by the main shaft	br.h.p.
d Turbo-generators, (including engine-driven generators)	One kilowatt-hour delivered at the bus-bar,† not including exciter output‡	kw-hr.
e Pumping Machinery	One gallon of water discharged to the force main in 24 hr	
	One gallon of water discharged per min.§	gal.pr.min.
	One water horsepower delivered to the force main, based on the total head including suction	w.h.p.
f Air Machinery	One cu. ft. of air at 62 deg. and 30 in.¶	cu.ft.
	One air horsepower	air h.p.
g Locomotives	One indicated horsepower developed in the main cylinders	i.h.p.
	One dynamometer horsepower delivered to the draw-bar	dyn. h.p.

* A subsidiary unit which may be used for stationary boilers is a "Boiler Horsepower," or 34½ lb. of water evaporated from and at 212 deg. per hour, i. e., from water at 212 deg. into steam at the same temperature. Electrical Engineers have suggested a unit termed "Myrowatt," which differs but little from boiler horsepower when expressed in B.t.u. per hour.

† It is assumed that the drop in voltage between generator terminals and switchboard is not over one-half of one per cent.

‡ If the exciter current is taken from an outside source the kw. thus supplied are to be deducted from the total output.

§ This unit applies to small pumps and some classes of large sized pumps.

¶ 30-in. barometer is referred to a temperature of 62 deg.; or 29.92-in. referred to 32 deg.

h Gas Producers.....	One pound of dry fuel consumed per hour.	
i Gas and Oil Engines.	One brake horsepower delivered by the main shaft.....	br.h.p.
j Waterwheels	One brake horsepower delivered by the main shaft.....	br.h.p.

TABLE 5 STANDARDS OF EFFICIENCY AND ECONOMY

a Boilers	Relation between B.t.u. absorbed by boiler per lb. of coal fired and calorific value of 1 lb. coal. (Efficiency of boiler furnace and grate.) Relation between B.t.u. absorbed by boiler, per lb. of combustible burned and calorific value of 1 lb. combustible. (Efficiency of boiler and furnace.)
b Reciprocating Steam Engines	(1) B.t.u. per i.h.p.-hr. (2) B.t.u. per brake h.p.-hr. (3) Thermal efficiency ratio referred to i.h.p. (4) Thermal efficiency ratio referred to br. h.p. (5) Lb. of dry steam per i.h.p.-hr. (6) Lb. of dry steam per br. h.p.-hr.
c Steam Turbines....	(1) B.t.u. per br. h.p.-hr. (2) Thermal efficiency ratio. (3) Lb. of dry steam per br. h.p.-hr.
d Turbo-generators, (including engine-driven generators)	(1) B.t.u. per kw-hr. (2) Thermal efficiency ratio. (3) Lb. of dry steam per kw-hr.
e Pumping Engines...	(1) Ft-lb. of work per million B.t.u. (2) Ft-lb. of work per 1000 lb. dry steam. (3) Ft-lb. of work per 100 lb. dry fuel.
f Air Machinery.....	(1) B.t.u. per net air h.p. per hr. (2) Lb. of dry steam per net air h.p. per hr. (3) Lb. of dry steam per 1000 cu. ft. of free air compressed to 100 lb. gage pressure reduced to atmospheric temperature.
g Locomotives	(1) Lb. of dry fuel per i.h.p.-hr. (2) Lb of dry fuel per dyn. h.p.-hr. (3) Lb. of dry steam per i.h.p.-hr. (4) Lb. of dry steam per dyn. h.p.-hr. (5) Lb. of dry fuel per ton-mile.
h Gas Producers.....	Relation between B.t.u. of the gas output per lb. of fuel charged and calorific value of 1 lb. of fuel.
i Gas and Oil Engines.	(1) B.t.u. per brake h.p.-hr. (2) Thermal efficiency ratio. (3) Lb. of dry fuel per br. h.p.-hr. (4) Cu. ft. of gas per br. h.p.-hr.
j Waterwheels	Relation between brake h.p. and potential h.p. of total water used.

k Steam Power Plants.	(1) Lb. of dry fuel per i.h.p.-hr., main engine.
	(2) Lb. of dry fuel per i.h.p.-hr., auxiliaries.
	(3) Lb. of dry fuel per i.h.p.-hr., whole plant.
	(4) Lb. of dry steam per i.h.p.-hr., main engine.
	(5) Lb. of dry steam per i.h.p.-hr., auxiliaries.
	(6) Lb. of dry steam per i.h.p.-hr., whole plant
l Electric Power Plants	(1) Lb. of dry fuel per kw-hr., main engine or turbine
	(2) Lb. of dry fuel per kw-hr., auxiliaries.
	(3) Lb. of dry fuel per kw-hr., whole plant.
	(4) Lb. of dry steam per kw-hr., main engine or turbine.
	(5) Lb. of dry steam per kw-hr., auxiliaries.
	(6) Lb. of dry steam per kw-hr., whole plant.
m Pumping Engine Plants	(1) Ft-lb. of work per million B.t.u.
	(2) Ft-lb. of work per 100 lb. dry fuel.
	(3) Ft-lb. of work per 1000 lb. dry steam.
n Gas Power Plants..	(1) Lb. of dry fuel per brake h.p.-hr.
	(2) Cu. ft. of gas per br. h.p.-hr.

The i.h.p. and brake h.p. in this table refer to that of the main engine, turbine, or waterwheel, and the kw. to the current measured at the bus-bar, not including exciter current.

Contracts for power plant apparatus should specify the leading dimensions of the apparatus and its rated capacity, expressed in the units given in Table 4. If a specific guarantee of capacity is made, either working capacity or maximum capacity, the operating conditions under which the guarantee is to be met should be clearly set forth; such, for example, as steam pressure, speed, vacuum, quality of fuel, force of draft, etc. Likewise if a contract contains a guarantee of economy all the conditions should be fully specified.

The commercial rating of capacity determined on for power plant apparatus, whether for the purpose of contracts for sale, or otherwise, should be such that a sufficient reserve capacity beyond the rating is available to meet the contingencies of practical operation; such contingencies, for example, as the loss of steam pressure and capacity due to cleaning fires, inferior coal, oversight of the attendants, sudden demand for an unusual output of steam or power, etc. To secure this end, the following requirements should be met:

(*a*) *Boilers.* The reserve capacity of a boiler should be at least one-third of the commercial rating, when using coal which is regarded as a standard where the boiler is located, the fire being crowded, and the draft between the damper and the boiler being at least ½-in. water column.

A sufficient amount of grate surface should be provided in such a boiler to develop the rated capacity with the coal and draft named without crowding the fire.

(*b*) *Reciprocating Steam Engines and Steam Turbines.* The reserve capacity of a steam engine or steam turbine at a stipulated steam pressure should be such as to allow a drop of at least 15 per cent in the pressure without material reduction in the normal speed. It should also allow an overload at the specified pressure amounting to at least 25 per cent of the rated power.

(*c*) *Pumping Engines.* The reserve capacity of a pumping engine should be such as to permit a drop in the steam pressure of at least 15 per cent without sensible reduction in the quantity of water discharged at its rated capacity, and to allow an increase in power sufficient to discharge 20 per cent more water than the rated amount.

(*d*) *Gas Producers.* The reserve capacity of a gas producer should be such that when forced it will burn in a given time 20 per cent more coal of the quality agreed upon than the rated capacity.

(*e*) *Gas and Oil Engines.* The reserve capacity of an internal-combustion engine should be such that when supplied with gas of the kind and quality which it is designed to use, it should develop at least 20 per cent more power than the commercial rating.

(*f*) *Waterwheels.* The reserve capacity of a waterwheel should be at least 10 per cent more than the commercial rating at the specified head, the buckets in the wheel being clean and the flow of water unobstructed.

PART III

RULES FOR SAMPLING AND DRYING COAL AND ASH AND SAMPLING STEAM

RULES FOR SAMPLING AND DRYING COAL AND ASH AND SAMPLING STEAM

(*A*) *Sampling and Drying Coal*

Select a representative shovelful from each barrow-load as it is drawn from the coal pile or other source of supply, and store the samples in a cool place in a covered metal receptacle. When all the coal has thus been sampled, break up the lumps, thoroughly mix the whole quantity, and finally reduce it by the process of repeated quartering and crushing to a sample weighing about 5 lb., the largest pieces being about the size of a pea. From this sample two 1-qt. air-tight glass fruit jars, or other air-tight vessels, are to be promptly filled and preserved for subsequent determinations of moisture, calorific value, and chemical composition. These operations should be conducted where the air is cool and free from drafts

When the sample lot of coal has been reduced by quartering to say 100 lb., a portion weighing say 15 to 20 lb. should be withdrawn for the purpose of immediate moisture determination. This is placed in a shallow iron pan and dried on the hot iron boiler flue for at least 12 hours, being weighed before and after drying on scales reading to quarter ounces.

The moisture thus determined is approximately reliable for anthracite and semi-bituminous coals, but not for coals containing much inherent moisture. For such coals, and for all absolutely reliable determinations the method to be pursued is as follows:

> Take one of the samples contained in the glass jars, and subject it to a thorough air drying, by spreading it in a thin layer and exposing it for several hours to the atmosphere of a warm room, weighing it before and after, thereby determining the quantity of surface moisture it contains. Then crush the whole of it by running it through an ordinary coffee mill or other suitable crusher adjusted so as to produce somewhat coarse grains (less than 1-16 in.), thoroughly mix the crushed sample, select from it a portion of from 10 to 50 grams[1], weigh it in a balance which will easily show a variation as small as 1 part in 1000, and dry it for one hour in an air or

[1] Say ½ oz. to 2 oz.

sand bath at a temperature between 240 and 280 deg. fahr. Weigh it and record the loss, then heat and weigh again until the minimum weight has been reached. The difference between the original and the minimum weight is the moisture in the air-dried coal. The sum of the moisture thus found and that of the surface moisture is the total moisture.

(*B*) *Sampling Ashes and Refuse*

The method above described may also be followed for obtaining a sample of the ashes and refuse.

(*C*) *Sampling Steam*

Construct a sampling pipe or nozzle made of ½-in. iron pipe and insert it in the steam main at a point where the entrained moisture is likely to be most thoroughly mixed. The inner end of the pipe, which should extend nearly across to the opposite side of the main, should be closed and the interior portion perforated with not less than twenty ⅛-in. holes equally distributed from end to end and preferably drilled in irregular or spiral rows, with the first hole not less than half an inch from the wall of the pipe.

The sampling pipe should not be placed near a point where water may pocket or where such water may affect the amount of moisture contained in the sample. Where non-return valves are used, or where there are horizontal connections leading from the boiler to a vertical outlet, water may collect at the lower end of the uptake pipe and be blown upward in a spray which will not be carried away by the steam owing to a lack of velocity. A sample taken from the lower part of this pipe will show a greater amount of moisture than a true sample. With goose-neck connections a small amount of water may collect on the bottom of the pipe near the upper end where the inclination is such that the tendency to flow backward is ordinarily counterbalanced by the flow of steam forward over its surface; but when the velocity momentarily decreases the water flows back to the lower end of the goose-neck and increases the moisture at that point, making it an undesirable location for sampling. In any case it should be borne in mind that with low velocities the tendency is for drops of entrained water to settle to the bottom of the pipe, and to be temporarily broken up into spray whenever an abrupt bend or other disturbance is met.

If it is necessary to attach the sampling nozzle at a point near the end of a long horizontal run, a drip pipe should be provided a short distance in front of the nozzle, preferably at a pocket formed by some fitting and the water running along the bottom of the main drawn off, weighed, and added to the moisture shown by the calori-

meter; or, better, a steam separator should be installed at the point noted.

(D) General Location of Sampling Pipe and Thermometer-Well

In testing a stationary boiler the sampling pipe should be located as near as practicable to the boiler, and the same is true as regards the thermometer well when the steam is superheated. In an engine or turbine test these locations should be as near as practicable to the throttle valve. In the test of a plant where it is desired to get complete information, especially where the steam main is unusually long, sampling nozzles or thermometer wells should be provided at both points, so as to obtain data at either point as may be required.

In a locomotive, the calorimeter should be attached either to the steam dome where it may be connected to the throttle opening, or to the steam passage in the saddle casting on one side.

SECOND SECTION: INDIVIDUAL CODES

PART IV

BOILER CODE

RULES FOR CONDUCTING EVAPORATIVE TESTS OF BOILERS

1 OBJECT AND PREPARATIONS

Determine the object, take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given on pages 1689 to 1714, and make preparations for the test accordingly.

2 FUEL

Determine the character of fuel to be used.[1] For tests of maximum efficiency or capacity of the boiler to compare with other boilers, the coal should be of some kind which is commercially regarded as a standard for the locality where the test is made.

In the Eastern States the standards thus regarded for semi-bituminous coals are Pocahontas (Va. and W. Va.) and New River (W. Va.); for anthracite coals those of the No. 1 buckwheat size, fresh-mined, containing not over 13 per cent ash by analysis; and for bituminous coals, Youghiogheny and Pittsburg coals. In some sections east of the Allegheny Mountains the semi-bituminous Clearfield (Pa.) and Cumberland (Md.) are also considered as standards. These coals when of good quality possess the essentials of excellence, adaptability to various kinds of furnaces, grates, boilers, and methods of firing required, besides being widely distributed and generally accessible in the Eastern market.

There are no special grades of coal mined in the Western States which are widely and generally considered as standards for testing purposes; the best coal obtainable in any particular locality being regarded as the standard of comparison.

A coal selected for maximum efficiency and capacity tests, should be the best of its class, and especially free from slagging and unusual clinker-forming impurities.

For guarantee and other tests with a specified coal containing not more than a certain amount of ash and moisture, the coal selected

[1] This code relates primarily to tests made with coal. For reference to oil and gas fuel tests see Section 17, Page 1725.

should not be higher in ash and in moisture than the stated amounts, because any increase is liable to reduce the efficiency and capacity more than the equivalent proportion of such increase.

The size of the coal, especially where it is of the anthracite class, should be determined by screening a suitable sample. (See page 1694 and Appendix No. 7 for list of sizes.)

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for boiler tests are:

(*a*) Platform scales for weighing coal and ashes
(*b*) Graduated scales attached to the water glasses
(*c*) Tanks and platform scales for weighing water (or water meters calibrated in place)
(*d*) Pressure gages, thermometers, and draft gages
(*e*) Calorimeters for determining the calorific value of fuel and the quality of steam
(*f*) Furnace pyrometers
(*g*) Gas analyzing apparatus

Full directions regarding the use and calibration of the above-mentioned appliances are given under the heading Apparatus and Instruments, pages 1691 to 1702, and on pages 1713 and 1714.

For particulars regarding the best location of the various instruments and apparatus, see Appendix No. 24.

4 OPERATING CONDITIONS

Determine what the operating conditions and method of firing should be to conform to the object in view, as pointed out on page 1703, and see that they prevail throughout the trial, as nearly as possible.

Where uniformity in the rate of evaporation is required, arrangement can usually be made to dispose of the steam so that this result can be attained. In a single boiler it may be accomplished by discharging steam through a waste pipe and regulating the amount by means of a valve. In a battery of boilers, in which only one is tested, the draft may be regulated on the remaining boilers to meet the varying demands for steam, leaving the test boiler to work under a steady rate of evaporation.

5 DURATION

The duration of tests to determine the efficiency of a hand-fired boiler, should be 10 hours of continuous running, or such time as

may be required to burn a total of 250 lb. of coal per square foot of grate.

In the case of a boiler using a mechanical stoker, the duration, where practicable, should be at least 24 hours. If the stoker is of a type that permits the quantity and condition of the fuel bed at beginning and end of the test to be accurately estimated, the duration may be reduced to 10 hours, or such time as may be required to burn the above noted total of 250 lb. per sq. ft.

In commercial tests where the service requires continuous operation night and day, with frequent shifts of firemen, the duration of the test, whether the boilers are hand-fired or stoker-fired, should be at least 24 hours. Likewise in commercial tests, either of a single boiler or of a plant of several boilers, which operate regularly a certain number of hours and during the balance of the day the fires are banked, the duration should not be less than 24 hours.

The duration of tests to determine the maximum evaporative capacity of a boiler, without determining the efficiency, should not be less than three hours.

6 STARTING AND STOPPING

The conditions regarding the temperature of the furnace and boiler, the quantity and quality of the live coal and ash on the grates, the water level, and the steam pressure, should be as nearly as possible the same at the end as at the beginning of the test.

To secure the desired equality of conditions with hand-fired boilers, the following method should be employed:

The furnace being well heated by a preliminary run, burn the fire low, and thoroughly clean it, leaving enough live coal spread evenly over the grate (say two to four in.)[1], to serve as a foundation for the new fire. Note quickly the thickness of the coal bed as nearly as it can be estimated or measured; also the water level[2], the steam pressure, and the time, and record the latter as the starting time. Fresh coal should then be fired from that weighed for the test, the ashpit thoroughly cleaned, and the regular work of the test proceeded with.

Before the end of the test the fire should again be burned low and cleaned in such a manner as to leave the same amount of live coal on the grate as at the start. When this condition is reached, observe quickly the water level[2], the steam pressure, and the time, and record the latter as the stopping time. If the water level is not

[1] 1 to 2 in. for small anthracite coals.

[2] Do not blow down the water-glass column for at least one hour before these readings are taken. An erroneous indication may otherwise be caused by a change of temperature and density of the water within the column and connecting pipe.

the same as at the beginning, a correction should be made by computation, rather than by feeding additional water after the final readings are taken. Finally remove the ashes and refuse from the ashpit.

In a plant containing several boilers where it is not practicable to clean them simultaneously, the fires should be cleaned one after the other as rapidly as may be, and each one after cleaning charged with enough coal to maintain a thin fire in good working condition. After the last fire is cleaned and in working condition, burn all the fires low (say 4 to 6 in.), note quickly the thickness of each, also the water levels, steam pressure, and time, which last is taken as the starting time. Likewise when the time arrives for closing the test, the fires should be quickly cleaned one by one, and when this work is completed they should all be burned low the same as at the start, and the various observations made as noted.

In the case of a large boiler having several furnace doors requiring the fire to be cleaned in sections one after the other, the above directions pertaining to starting and stopping in a plant of several boilers may be followed.

To obtain the desired equality of conditions of the fire when a mechanical stoker other than a chain grate is used, the procedure should be modified where practicable as follows :

Regulate the coal feed so as to burn the fire to the low condition required for cleaning. Shut off the coal-feeding mechanism and fill the hoppers level full. Clean the ash or dump plate, note quickly the depth and condition of the coal on the grate, the water level[1], the steam pressure, and the time, and record the latter as the starting time. Then start the coal-feeding mechanism, clean the ashpit, and proceed with the regular work of the test.

When the time arrives for the close of the test, shut off the coal-feeding mechanism, fill the hoppers and burn the fire to the same low point as at the beginning. When this condition is reached, note the water level, the steam pressure, and the time, and record the latter as the stopping time. Finally clean the ash plate and haul the ashes.

In the case of chain grate stokers, the desired operating conditions should be maintained for half an hour before starting a test and for a like period before its close, the height of the throat plate and the speed of the grate being the same during both of these periods.

7 RECORDS

The records of data should be obtained as pointed out in Part I, page 1703. Half-hourly readings of the instruments are usually sufficient. If there are sudden and wide fluctuations, the readings in such cases should be taken every fifteen minutes, and in some instances oftener.

The coal should be weighed and delivered to the firemen in portions suf-

[1] See footnote, p. 1719.

ficient for one hour's run, thereby ascertaining the degree of uniformity of firing. An ample supply of coal should be maintained at all times, but the quantity on the floor at the end of each hour should be as small as practicable, so that the same may be readily estimated and deducted from the total weight.

The records should be such as to ascertain also the consumption of feedwater each hour, and thereby determine the degree of uniformity of evaporation.

8 QUALITY OF STEAM

If the boiler does not produce superheated steam the percentage of moisture in the steam should be determined by the use of a throttling or separating calorimeter, in the manner pointed out in Part III, page 1713. If the boiler has superheating surface, the temperature of the steam should be determined by the use of a thermometer inserted in a thermometer well, as pointed out on page 1696, Part I.

9 SAMPLING AND DRYING COAL

During the progress of the test the coal should be regularly sampled for the purpose of analysis and determination of moisture, in the manner pointed out in Part III, page 1712.

10 ASHES AND REFUSE

The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed so far as possible in a dry state. If wet the amount of moisture should be ascertained and allowed for, a sample being taken and dried for this purpose. This sample may serve also for analysis and the determination of unburned carbon and fusing temperature.

11 CALORIFIC TESTS AND ANALYSES OF COAL

The quality of the fuel should be determined by calorific tests and analysis of the coal sample above referred to. Directions for making these tests and analyses will be found in Part I under the headings Coal Calorimeters and Coal Analysis Apparatus, pages 1697 and 1698 and in Appendices Nos. 12 and 14.

12 ANALYSES OF FLUE GASES

For approximate determinations of the composition of the flue

gases, the Orsat apparatus, or some modification thereof, should be employed. If momentary samples are obtained the analyses should be made as frequently as possible, say every 15 to 30 minutes, depending on the skill of the operator, noting at the time the sample is drawn the furnace and firing conditions. If the sample drawn is a continuous one, the intervals may be made longer.

Fuller directions will be found in Part I, under the heading Gas Analysis Apparatus, page 1698, and in Appendix No. 15.

13 SMOKE OBSERVATIONS

In tests of bituminous coals requiring a determination of the amount of smoke produced, observations should be made regularly throughout the trial at intervals of five minutes (or if necessary every minute), noting at the same time the furnace and firing conditions.

Full particulars regarding these observations are given in Part I, under the heading Appliances and Methods Pertaining to Smoke Determination, page 1698, and in Appendices Nos. 16 and 17.

14 CALCULATION OF RESULTS

The methods to be followed in expressing and calculating those results which are not self-evident are explained as follows:

(*a*) *Efficiency.* The "efficiency of boiler, furnace and grate" is the relation between the heat absorbed per pound of coal fired, and the calorific value of one pound of coal.

The "efficiency of boiler and furnace" is the relation between the heat absorbed per pound of combustible burned, and the calorific value of one pound of combustible. This expression of efficiency furnishes a means for comparing one boiler and furnace with another, when the losses of unburned coal due to grates, cleanings, etc., are eliminated.

The "combustible burned" is determined by subtracting from the weight of coal supplied to the boiler, the moisture in the coal, the weight of ash and unburned coal withdrawn from the furnace and ashpit, and the weight of dust, soot, and refuse, if any, withdrawn from the tubes, flues, and combustion chambers, including ash carried away in the gases, if any, determined from the analyses of coal and ash. The "combustible" used for determining the calorific value is the weight of coal less the moisture and ash found by analysis.

The "heat absorbed" per pound of coal, or combustible, is calculated by multiplying the equivalent evaporation from and at 212 deg. per lb. of coal or combustible by 970.4.

(*b*) *Corrections for Moisture in Steam.* When the percentage is less than 2 per cent it is sufficient merely to deduct the percentage from the weight of water fed. If the percentage is greater than 2 per cent or if extreme accuracy is required, the factor of correction equals

$$Q + P\frac{(T - h)}{(H - h)}$$

in which Q is the quality of the steam (one minus the decimal epresenting the percentage of moisture), P the proportion of moisture, T the total heat of water at the temperature of the steam, h the total heat of the feedwater, and H the total heat of saturated steam of the given temperature.

(*c*) *Correction for live steam, if any, used for aiding Combustion.* If live steam is admitted into the furnace or ash-pit for producing blast, injecting fuel, or aiding combustion, it is to be deducted from the total evaporation, and the net evaporation used in the various calculations.

(*d*) *Equivalent Evaporation.* The equivalent evaporation from and at 212 deg. is obtained by multiplying the weight of water evaporated, corrected for moisture in steam, by the "factor of evaporation." The latter equals

$$\frac{H - h}{970.4}$$

in which H and h are respectively the total heat of saturated steam and of the feedwater entering the boiler. When the steam is superheated, the total heat of the steam is that of saturated steam plus the product of the number of degrees of superheating by the specific heat of the steam.

Unless otherwise provided, a combined boiler and superheater should be treated as one unit, and the equivalent of the work done by the superheater should be included in the evaporative work of the boiler.

(*e*) *Heat Balance.* The "heat balance," or approximate distribution of the calorific value of the coal or combustible among the several items of heat utilized and heat lost, should be obtained in cases where the flue gases have been analyzed and a complete analysis made of the coal.

The loss due to moisture in the coal is found by multiplying the total heat of one pound of superheated steam at the temperature of the escaping gases, calculated from the temperature of the air in the boiler room, by the proportion of moisture.

The loss due to moisture formed by the burning of hydrogen is obtained by multiplying the total heat of one pound of superheated steam at the temperature of the escaping gases, calculated from the temperature of the air in the boiler room, by the proportion of the hydrogen, determined from the analysis of the coal, and multiplying the result by 9.

The loss due to heat carried away in the dry gases is found by

multiplying the weight of gas per pound of coal or combustible by the elevation of temperature of the gases above the temperature of the boiler room, and by the specific heat of the gases (0.24). The weight of gas referred to is obtained by finding the weight of dry gas per pound of carbon burned, using the formula

$$\frac{11\,CO_2 + 8\,O + 7\,(CO + N)}{3\,(CO_2 + CO)}$$

in which CO_2, CO, O, and N are expressed in percentages by volume, and multiplying this result by the proportion borne by the carbon burned to the whole amount of coal or combustible as determined from the results of the analysis of the coal, ash, and refuse.

The loss due to incomplete combustion of carbon is found by first obtaining the proportion borne by the carbon monoxide in the gases to the sum of the carbon monoxide and carbon dioxide, and then multiplying this proportion by the proportion of carbon in the coal or combustible, and finally multiplying the product by 10,150, which is the number of heat units generated by burning to carbon dioxide one pound of carbon contained in carbon monoxide.

The loss due to combustible matter in the ash and refuse is found by multiplying the proportion that this combustible bears to the whole amount of coal or combustible, by its calorific value per pound. For most purposes it is sufficient to assume the latter to be 14,600 B.t.u., the same as that of carbon.

The loss due to moisture in the air is determined by multiplying the weight of such moisture per pound of coal or combustible by the elevation of temperature of the flue gases above the temperature of the boiler room and by 0.47. The weight of moisture is found by multiplying the weight of air per pound of coal or combustible by the moisture in one pound of air determined from readings of the wet and dry-bulb thermometer.

(*f*) *Total Heat of Combustion of Coal, by Analysis.* The total heat of combustion may be computed from the results of the ultimate analysis by using the formula

$$14{,}600\,C + 62{,}000\,(H - \frac{O}{8}) + 4000\,S,$$

in which C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur, respectively.

(*g*) *Air for Combustion.* The quantity of air used may be calculated by the formulae:

$$\text{Lb. of air per lb. of carbon} = \frac{3.032\,N}{CO_2 + CO}$$

in which N, CO_2 and CO are the percentages of dry gas obtained by analysis, and

Lb. of air per lb. of coal = lb. air per lb. C × per cent C in the coal.

The ratio of the air supply to that theoretically required for complete combustion is $\frac{N}{N - 3.782\,O}$.

15 DATA AND RESULTS

The data and results should be reported in accordance with either the Short Form or the Complete Form printed below, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

16 CHART

In trials having for an object the determination and exposition of the complete boiler performance, the entire log of readings and data should be plotted on a chart and represented graphically. See Appendix No. 23.

17 TESTS WITH OIL AND GAS FUELS

Tests of boilers using oil or gas for fuel should accord with the rules here given, excepting as they are varied to conform to the particular characteristics of the fuel. The duration in such cases may be reduced, and the "flying" method of starting and stopping employed.

The table of data and results should contain items stating character of furnace and burner, quality and composition of oil or gas, temperature of oil, pressure of steam used for vaporizing and quantity of steam used for both vaporizing and for heating.

TABLE 1 DATA AND RESULTS OF EVAPORATIVE TEST SHORT FORM. CODE OF 1912

(1)	Test of.....boiler located at..... to determine....., conducted by.....	
(2)	Kind of furnace.....	
(3)	Grate surface.....	sq. ft.
(4)	Water-heating surface.....	sq. ft.
(5)	Superheating surface.....	sq. ft.
(6)	Date	
(7)	Duration	hr.
(8)	Kind and size of coal.....	
	AVERAGE PRESSURES, TEMPERATURES, ETC.	
(9)	Steam pressure by gage.....	lb.
(10)	Temperature of feedwater entering boiler.....	deg.
(11)	Temperature of escaping gases leaving boiler.....	deg.
(12)	Force of draft between damper and boiler.....	in.
(13)	Percentage of moisture in steam, or number deg. of superheating	per cent or deg.

Total Quantities

(14) Weight of coal as fired[1] ..... lb.
(15) Percentage of moisture in coal ..... per cent.
(16) Total weight of dry coal consumed ..... lb.
(17) Total ash and refuse ..... lb.
(18) Percentage of ash and refuse in dry coal ..... per cent.
(19) Total weight of water fed to the boiler[2] ..... lb.
(20) Total water evaporated, corrected for moisture in steam ..... lb.
(21) Total equivalent evaporation from and at 212 deg ..... lb.

Hourly Quantities and Rates

(22) Dry coal consumed per hour ..... lb.
(23) Dry coal per sq. ft. of grate surface per hour ..... lb.
(24) Water evaporated per hour corrected for quality of steam ..... lb.
(25) Equivalent evaporation per hour from and at 212 deg ..... lb.
(26) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water-heating surface ..... lb.

Capacity

(27) Evaporation per hour from and at 212 deg. (same as Line 25) .... lb.
(28) Boiler horsepower developed (Item 27 ÷ 34½) ..... bl.h.p.
(29) Rated capacity, in evaporation from and at 212 deg. per hour ..... lb.
(30) Rated boiler horsepower ..... bl.h.p.
(31) Percentage of rated capacity developed ..... per cent.

Economy Results

(32) Water fed per lb. of coal fired (Item 19 ÷ Item 14) ..... lb.
(33) Water evaporated per lb. of dry coal (Item 20 ÷ Item 16) ..... lb.
(34) Equivalent evaporation from and at 212 deg. per lb. of dry coal (Item 21 ÷ Item 16) ..... lb.
(35) Equivalent evaporation from and at 212 deg. per lb. of combustible [Item 21 ÷ (Item 16 — Item 17)] ..... lb.

Efficiency

(36) Calorific value of 1 lb. of dry coal ..... B.t.u.
(37) Calorific value of 1 lb. of combustible ..... B.t.u.
(38) Efficiency of boiler, furnace and grate $\left[100 \times \frac{\text{Item } 34 \times 970.4}{\text{Item } 36}\right]$ per cent.
(39) Efficiency of boiler and furnace $\left[100 \times \frac{\text{Item } 35 \times 970.4}{\text{Item } 37}\right]$ .... per cent.

Cost of Evaporation

(40) Cost of coal per ton of .... lb. delivered in boiler room ..... dollars
(41) Cost of coal required for evaporating 1000 lb. of water from and at 212 deg ..... dollars

[1] The term "as fired" means actual condition including moisture, corrected for estimated difference in weight of coal on the grate at beginning and end.

[2] Corrected for inequality of water level and steam pressure at beginning and end.

TABLE 2 DATA AND RESULTS OF EVAPORATIVE TEST COMPLETE FORM, CODE OF 1912

(1) Test of.....boiler located at.....
to determine....., conducted by.....

Dimensions, Proportions, etc.

(2) Number and kind of boilers.....
(3) Kind of furnace.....
(4) Grate surface.....width.....length.....area.....sq. ft.
(5) Approximate width of air spaces in grate.....in.
(6) Proportion of air space to whole grate surface.....per cent.
(7) Water-heating surface.....sq. ft.
(8) Superheating surface.....sq. ft.
(9) Ratio of water-heating surface to grate surface.....to 1
(10) Ratio of minimum draft area to grate surface.....1 to....
(11) Date .....
(12) Duration .....hr.
(13) Kind of coal.....
(14) Size of coal.....

Average Pressures, Temperatures, etc.

(15) Steam pressure by gage.....lb.
(16) Barometric pressure.....lb.
(17) Force of draft at dampers of individual boilers.....in.
(18) Force of draft in main flue near boilers.....in.
(19) Force of draft in main flue between economizer and chimney....in.
(20) Force of draft in furnaces.....in.
(21) Force of blast in ashpits.....in.
(22) State of weather.....
(23) Temperature of external air.....deg.
(24) Temperature of fireroom.....deg.
(25) Temperature of steam.....deg.
(26) Normal temperature of saturated steam.....deg.
(27) Temperature of feedwater entering flue heater or economizer..deg.
(28) Temperature of feedwater leaving heater or economizer and entering boilers.....deg.
(29) Temperature of gases leaving boilers.....deg.
(30) Temperature of gases leaving economizer.....deg.

Quality of Steam.

(31) Percentage of moisture in steam.....per cent
(32) Number of degrees of superheating.....deg.
(33) Quality of steam (dry steam = unity).....

Total Quantities.

(34) Weight of coal as fired[1].....lb.
(35) Percentage of moisture in coal.....per cent.
(36) Total weight of dry coal consumed.....lb.
(37) Total ash and refuse.....lb.

(38) Total combustible consumed (Line 36—Line 37).....lb.
(39) Percentage of ash and refuse in dry coal.....per cent
(40) Total weight of water fed to boiler.....lb.
(41) Total water evaporated corrected for moisture in steam.....lb.
(42) Factor of evaporation, based on temperature of water entering boilers .....
(43) Total equivalent evaporation from and at 212 deg.....lb.

HOURLY QUANTITIES AND RATES.

(44) Dry coal consumed per hour.....lb.
(45) Combustible consumed per hour.....lb.
(46) Dry coal per sq. ft. of grate surface per hour.....lb.
(47) Water evaporated per hour, corrected for quality of steam.....lb.
(48) Equivalent evaporation per hour from and at 212 deg.[3].....lb.
(49) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water-heating surface[3].....lb.

PROXIMATE ANALYSIS OF COAL.

(50) Fixed carbon.....per cent
(51) Volatile matter.....per cent
(52) Moisture .....per cent
(53) Ash .....per cent

100 per cent

(54) Sulphur, separately determined.....per cent

ULTIMATE ANALYSIS OF DRY COAL.

(55) Carbon (C).....per cent
(56) Hydrogen (H).....per cent
(57) Oxygen (O).....per cent
(58) Nitrogen (N).....per cent
(59) Sulphur (S).....per cent
(60) Ash .....per cent

100 per cent

(61) Moisture in sample of coal as received.....per cent

ANALYSIS OF ASH AND REFUSE.

(62) Carbon .....per cent
(63) Earthy matter.....per cent
(*a*) SiO_2
(*b*) Al_2O_3 and Fe_2O_3
(*c*) MgO
(*d*) CaO
(64) Temperature of fushion of ash.....deg.

CALORIFIC VALUE.

(65) Calorific value of 1 lb. of dry coal by calorimeter.....B.t.u.
(66) Calorific value of 1 lb. of combustible by calorimeter.....B.t.u.
(67) Calorific value of 1 lb. of dry coal by analysis.....B.t.u.
(68) Calorific value of 1 lb. of combustible by analysis.....B.t.u.

CAPACITY.

(69) Evaporation per hour from and at 212 deg. (same as Line 48)....lb.
(70) Boiler horsepower developed (Line 69 ÷ 34½).....bl.h.p.
(71) Rated capacity per hour, from and at 212 deg.....lb.
(72) Rated boiler horsepower.....bl.h.p.
(73) Percentage of rated capacity developed.....per cent

ECONOMY RESULTS.

(74) Water fed per lb. of coal (Item 40 ÷ Item 34).....lb.
(75) Water evaporated per lb. of dry coal (Item 41 ÷ Item 36).....lb.
(76) Equivalent evaporation from and at 212 deg. per lb. of coal fired (Item 43 ÷ Item 34).....lb.
(77) Equivalent evaporation from and at 212 deg. per lb. of dry coal (Item 43 ÷ Item 36).....lb.
(78) Equivalent evaporation from and at 212 deg. per lb. of combustible (Item 43 ÷ Item 38.....lb.

EFFICIENCY.

(79) Efficiency of boiler, furnace, and grate $\left[100 \times \frac{\text{Item } 77 \times 970.4}{\text{Item } 65}\right]$ per cent

(80) Efficiency of boiler and furnace $\left[100 \times \frac{\text{Item } 78 \times 970.4}{\text{Item } 66}\right]$ ..per cent

COST OF EVAPORATION.

(81) Cost of coal per ton of .... lb. delivered in boiler room....dollars
(82) Cost of coal required for evaporating 1000 lb. of water under observed conditions.....dollars
(83) Cost of coal required for evaporating 1000 lb. of water from and at 212 deg.....dollars

SMOKE DATA.

(84) Percentage of smoke as observed.....per cent
(85) Weight of soot per hour obtained from smoke meter.....

METHODS OF FIRING.

(86) Kind of firing, whether spreading, alternate, or cooking.....
(87) Average thickness of fire.....
(88) Average intervals between firings for each furnace during time when fires are in normal condition.....
(89) Average interval between times of leveling or breaking up.....

ANALYSIS OF DRY GASES BY VOLUME.

(90) Carbon dioxide (CO_2).....per cent
(91) Oxygen (O).....per cent
(92) Carbon monoxide (CO).....per cent
(93) Hydrogen and hydrocarbons.....per cent
(94) Nitrogen, by difference (N).....per cent

100 per cent

HEAT BALANCE, BASED ON DRY COAL AND COMBUSTIBLE

	Dry Coal as Fired		Combustible Burned	
	B.t.u.	Per Cent	B.t.u.	Per Cent
(95) Heat absorbed by the boiler (Line 76 or 77x970.4)				
(96) Loss due to evaporation of moisture in coal				
(97) Loss due to heat carried away by steam formed by the burning of hydrogen				
(98) Loss due to heat carried away in the dry flue gases				
(99) Loss due to carbon monoxide				
(100) Loss due to combustible in ash and refuse				
(101) Loss due to heating moisture in air				
(102) Loss due to unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for				
(103) Total calorific value of 1 lb. of dry coal or combustible (Lines 65 and 66)		100		100

[1] The term "as fired" means actual condition including moisture, corrected for estimated difference in weight of coal on the grate at beginning and end.

[2] Corrected for inequality of water level and of steam pressure at beginning and end.

[3] The symbol U. E. meaning Units of Evaporation, may be substituted for the expression Equivalent water evaporated into dry steam from and at 212 deg.

PART V

STEAM ENGINE CODE

RULES FOR CONDUCTING TESTS OF RECIPROCATING ENGINES

1 OBJECT AND PREPARATIONS

Determine the object, take the dimensions, note the physical conditions not only of the engine but of all parts of the plant that are concerned in the determinations, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given on pages 1689 to 1714, and prepare for the test accordingly.

2 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple performance test of a steam engine, in which the steam consumption is determined by feedwater measurement, are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place)
(*b*) Graduated scales attached to the water glasses of the boilers
(*c*) Pressure gages, vacuum gages, and thermometers
(*d*) A steam calorimeter
(*e*) A barometer
(*f*) Steam engine indicators
(*g*) A planimeter
(*h*) A tachometer or other speed measuring apparatus
(*i*) A friction brake or dynamometer

Directions regarding the use and calibration of these appliances are given on pages 1691 to 1702, and on page 1713.

The determination of the heat and steam consumption of an engine by feedwater test requires the measurement of the various supplies of water fed to the boiler; that of the water discharged by separators and drips not returned to the boiler, and that of water and steam which escapes by leakage of the boiler and piping; all of these last being deducted from the total feedwater measured.

Where a surface condenser is provided and the steam consumption is determined from the water discharged by the air pump, no such measurement of drips and leakage is required, but assurance must be had

that all the steam passing into the cylinders finds its way into the condenser. If the condenser leaks the defects causing it should be remedied, or suitable correction made for the leakage.

To ascertain the consumption of heat, the various feed temperatures are taken and heat calculations made accordingly. If the conditions imposed by the particular method adopted for carrying on the test depart from the usual practice, as for example where a colder supply of feedwater is used than the ordinary supply, a preliminary or subsequent run should be made to ascertain the temperatures which obtain under the usual working conditions, and the heat measurements obtained under the test-conditions appropriately corrected for such departures.

The steam consumed by steam-driven auxiliaries which are required for the operation of the engine should be included in the total steam from which the heat consumption is calculated and the quantity of steam thus used should be determined and reported.

For fuller particulars, see description of Steam Measuring apparatus, page 1692, Part I, and Appendices Nos. 3 and 25.

3 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial, as pointed out on page 1703, Part I.

4 DURATION

A test for heat or steam consumption, with substantially constant load, should be continued for such time as may be necessary to obtain a number of successive hourly records, during which the results are reasonably uniform. For a test involving the measurement of feedwater for this purpose, five hours is sufficient duration. Where a surface condenser is used, and the measurement is that of the water discharged by the air pump, the duration may be somewhat shorter. In this case, successive half-hourly records may be compared and the time correspondingly reduced.

When the load varies widely at different times of the day, the duration should be such as to cover the entire period of variation.

The preliminary or subsequent trial for determining the working temperatures on a heat test, where the temperatures obtained under the test conditions depart from the usual temperatures, should be of such duration as may be required to secure working results.

5 STARTING AND STOPPING

The engine and appurtenances having been set to work and thoroughly heated under the prescribed conditions of test, except in cases where the object is to obtain the performance under working conditions, note the water levels in the boilers and feed reservoir, take the time and consider this the starting time. Then begin the measurements and observations and carry them forward until the end of the period determined on. When this time arrives, the water levels and steam pressure should be brought as near as practicable to the same points as at the start. This being done, again note the time and consider it the stopping time of the test. If there are differences in the water levels, proper corrections are to be applied.

Where a surface condenser is used, the collection of water discharged by the air pump begins at the starting time, and the water is thereafter measured or weighed until the end of the test; no observations of the boilers being required.

Care should be taken in cases where the activity of combustion in the boiler furnaces affects the height of water in the gage glasses that the same conditions of fire and drafts are operating at the end as at the beginning. For this reason it is best to start and stop a test without interfering with the regularity of the operation of the feed pump, provided the latter can be regulated to run so as to supply the feedwater at a uniform rate. In some cases where the supply of feedwater is irregular, as, for example, where an injector is used of a larger capacity than is required, the supply of feedwater should be temporarily shut off.

Suitable care should be observed in noting the average height of the water in the glasses, taking sufficient time to satisfactorily judge of the full extent of the fluctuation of the water line, and thereby its mean position.

6 RECORDS

The general data should be recorded as pointed out in Part I, page 1703, under the heading Records. Half-hourly readings of the instruments are sufficient, excepting where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes it necessary. Mark on each card the cylinder and the end on which it was taken, also the time of day. Record on one card of each set the readings of the pressure gages concerned, taken at the same time. These records should subsequently be entered on the

general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are worked up.

Further directions may be found in Part I, page 1699, under the heading Indicators, and in Appendix No. 18.

7 CALCULATION OF RESULTS

(*a*) *Dry Steam.* The quantity of dry steam consumed when there is no superheating is determined by deducting the moisture found by calorimeter test from the total amount of feedwater (the latter being corrected for leakages) or from the amount of air pump discharge, as the case may be.

When there is superheating the dry steam is found by multiplying the weight of superheated steam by the factor

$$1 + \frac{C\,(T - t)}{H - h}$$

in which

C = specific heat of superheated steam at observed pressure and temperature.

T = temperature of superheated steam.

t = temperature of saturated steam.

H = total heat of saturated steam of observed pressure.

h = total heat of feedwater.

(*b*) *Heat Consumption.* The number of heat units consumed by the engine is found by multiplying the weight of feedwater consumed, corrected for leakages, by the total heat of the steam above the working feed temperature, and multiplying the product by a factor of correction expressing the quality of the steam.

If the steam contains moisture, this factor equals

$$Q + P\,\frac{T - h}{H - h}$$

in which Q is the quality of the steam (one minus the decimal representing the percentage of moisture), P the proportion of moisture, T the total heat of water at the temperature of the steam, h the total heat of the feedwater, and H the total heat of saturated steam.

If the steam is superheated, the factor is that given above under (*a*) Dry Steam.

If there are a number of sources of feedwater supply, the corresponding heat units should be determined for each supply and the various quantities added together.

The British standard of heat consumption is based on a feedwater temperature assumed to be that of the temperature of saturated steam corresponding to the observed back pressure (whether this is above or below the atmosphere), plus the temperature due to heat derived from jacket or reheater drips. It does not include the heat consumed by any auxiliaries, except jackets and reheaters.

For fuller particulars see Appendices No. 25 and No. 26.

(*c*) *Indicated Horsepower.* In a single double-acting cylinder the indicated horsepower is found by using the formula

$$\frac{P\ L\ A\ N}{33{,}000}$$

in which P represents the average mean effective pressure in pounds per sq. in. measured from the indicator diagrams, L the length of stroke in ft., A the area of the piston less one-half the area of the piston rod, or the mean area of the rod if it passes through both cylinder heads, in sq. in., and N the number of single strokes per minute.

Where extreme accuracy is required, the power developed by each side of the piston may be determined and the results added together.

(*d*) *Brake Horsepower.* The brake horsepower is found by multiplying the net weight on the brake arm (the gross weight minus the weight when the brake is entirely free) in lb., the circumference of the circle passing through the bearing point at the end of the brake arm, in ft., and the number of revolutions of the brake shaft per minute; and dividing the product by 33,000.

(*e*) *Electrical Horsepower.* The electrical horsepower for a direct-connected generator is found by dividing the output at the bus-bar, expressed in kilowatts, by the decimal 0.746. For alternating-current systems the net output is to be used, being the total output less that consumed for excitation. See Appendix No. 21.

(*f*) *Efficiency.* The efficiency is expressed by the thermal efficiency ratio, which is found by dividing the quantity 2545 by the number of heat units consumed per h.p.-hr., either indicated or brake.

See Appendix No. 13 for reference to British practice.

(*g*) *Steam accounted for by Indicator Diagrams.* The steam accounted for, expressed in pounds per i.h.p. per hour, may readily be found by using the formula

$$\frac{13{,}750}{\text{m.e.p.}}\left[\ (C + E)\,Wc - (H + E)\ Wh\ \right]$$

in which

m.e.p = mean effective pressure.
C = proportion of stroke completed at cut-off or release
E = proportion of clearance
H = proportion of stroke uncompleted at compression
Wc = weight of 1 cu. ft. steam at cut-off or release pressure
Wh = weight of 1 cu. ft. steam at compression pressure

The points of cut-off release and compression, referred to are indicated in Fig. 1.*

In multiple expansion engines the mean effective pressure to be used in the above formula is the combined m.e.p. referred to the cylinder under consideration. In a compound engine the combined

* Reproduced from Report of Committee on Standardizing Engine Tests, Fig. 122, Trans. Am. Soc. M. E., vol. 24, p. 744.

m.e.p. for the h.p. cylinder is the sum of the actual m.e.p. of the h.p. cylinder and that of the l.p. cylinder multiplied by the cylinder ratio. Likewise the combined m.e.p. for the l.p. cylinder is the sum of the actual m.e.p. of the l.p. cylinder and the m.e.p. of the h.p. cylinder divided by the cylinder ratio.

(*h*) *Cut-Off and Ratio of Expansion.* To find the percentage of cut-off, or what may best be termed the "commercial cut-off," the following rule should be observed:

Through the point of maximum pressure during admission draw a line parallel to the atmospheric line. Through a point on the expansion line where the cut-off is complete, draw a

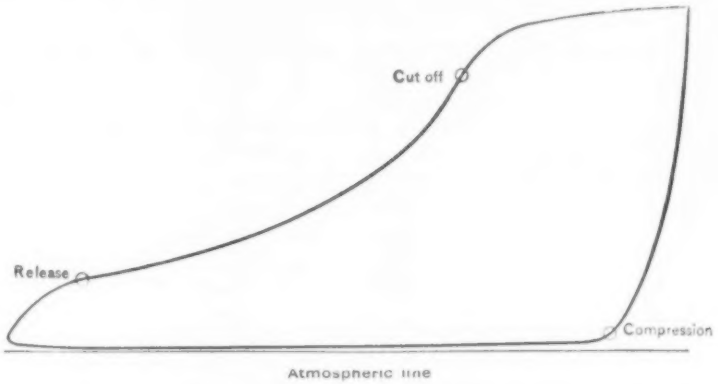

FIG. 1 POINTS WHERE "STEAM ACCOUNTED FOR BY INDICATOR" IS COMPUTED

hyperbolic curve. The intersection of these two lines is the point of commercial cut-off, and the proportion of cut-off is found by dividing the length measured on the diagram up to this point by the total length.

To find the ratio of expansion divide the volume corresponding to the piston displacement, including clearance, by the volume of the steam at the commercial cut-off, including clearance.

In a multiple expansion engine the ratio of expansion is found by dividing the volume of the l.p. cylinder, including clearance, by the volume of the h.p. cylinder at the commercial cut-off, including clearance.

(*i*) *Miscellaneous.* The method of obtaining a "combined diagram" in compound and other multiple expansion engines is pointed out in Appendix No. 27. That of finding the "diagram factor" is given in Appendix No. 28. Methods of making "heat analyses," and "temperature entropy diagrams," and the manner of determining the "ratio of economy of an engine to that of an ideal engine," are given in Appendices Nos. 29, 30, and 31, respectively.

8 DATA AND RESULTS

The data and results should be reported in accordance with either the Short Form or Complete Form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

(See footnote on page 1744 for reference to engines driving electric generators and other machinery.)

TABLE 1 DATA AND RESULTS OF HEAT AND FEEDWATER TESTS OF STEAM ENGINE

SHORT FORM, CODE OF 1912

		1st Cyl.	2d Cyl.	3d Cyl.
(1)	Test of.....engine located at.....			
	to determine.....conducted by.....			
(2)	Type and class of engine and auxiliaries.....			
(3)	Dimensions of main engine:			
	(*a*) Diameter of cylinder.....in.	.....	.....	.....
	(*b*) Stroke of piston.....ft.	.....	.....	.....
	(*c*) Diameter of piston rod each end, in	.....	.....	.....
	(*d*) Average clearance.....per cent	.....	.....	.....
	(*e*) Cylinder ratio.....	.....	.....	.....
	(*f*) Horsepower constant for 1 lb. m.e.p. and 1 r.p.m.....	.....	.....	.....
(4)	Dimensions and type of auxiliaries.....			
(5)	Date			
(6)	Duration ....hr.			

AVERAGE PRESSURES AND TEMPERATURES

(7)	Pressure in steam pipe near throttle by gage	lb.
(8)	Barometric pressure of atmosphere in in. of mercury	in.
(9)	Pressure in receivers by gage	lb.
(10)	Vacuum in condenser in in. of mercury	in.
(11)	Pressure in jackets and reheaters by gage	lb.
(12)	Temperature of main supply of feedwater	deg.
(13)	Temperature of additional supplies of feedwater	deg.

TOTAL QUANTITIES

(14)	Total water fed to boilers from main source of supply	lb.
(15)	Total water fed from additional supplies	lb.
(16)	Total water fed to boilers from all sources	lb.
(17)	Moisture in steam or superheating near throttle	per cent or deg.
(18)	Factor of correction for quality of steam	
(19)	Total dry steam consumed for all purposes	lb.

Hourly Quantities

(20) Water fed from main source of supply.....lb.
(21) Water fed from additional supplies.....lb.
(22) Total water fed to boilers per hour.....lb.
(23) Total dry steam consumed per hour.....lb
(24) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant.....lb.
(25) Net dry steam consumed per hour by engine and auxiliaries...lb.
(26) Net dry steam consumed per hour:
  (*a*) By engine alone.....lb.
  (*b*) By auxiliaries.....lb.

Heat Data

(27) Heat units per lb. of dry steam, based on temperature of Line 12.....B.t.u.
(28) Heat units per lb. of dry steam, based on temperature of Line 13.....B.t.u.
(29) Heat units consumed per hour, main supply of feed.....B.t.u.
(30) Heat units consumed per hour, additional supplies of feed....B.t.u.
(31) Total heat units consumed per hour for all purposes.....B.t.u.
(32) Loss of heat per hour due to leakage of plant, drips, etc.....B.t.u.
(33) Net heat units consumed per hour:
  (*a*) By engine and auxiliaries.....B.t.u.
  (*b*) By engine alone.....B.t.u.
  (*c*) By auxiliaries.....B.t.u.

Indicator Diagrams

		1st Cyl.	2d Cyl.	3d Cyl.
(34)	Commercial cut-off in per cent of stroke..	.....	.....	.....
(35)	Initial pressure in lb. per sq. in. above atmosphere	.....	.....	.....
(36)	Back pressure at lowest point above or below atmosphere in lb. per sq. in.....	.....	.....	.....
(37)	Mean effective pressure in lb. per sq. in...	.....	.....	.....
(38)	Steam accounted for by indicator in lb. per i.h.p. per hour:			
	(*a*) Near cut-off.....	.....	.....	.....
	(*b*) Near release.....	.....	.....	.....

Speed

(39) Revolutions per minute.....rev.
(40) Piston speed in ft. per min.....ft.

Power

(41) Indicated horsepower developed by main-engine cylinders:
  1st cylinder.....i.h.p.
  2d cylinder.....i.h.p.
  3d cylinder whole engine.....i.h.p.
  Whole engine.....i.h.p.
(42) Brake horsepower.....br.h.p.

ECONOMY RESULTS

(43) Heat units consumed by engine and auxiliaries per hour:
  (*a*) Per indicated horsepower.....B.t.u.
  (*b*) Per brake horsepower.....B.t.u.
(44) Dry steam consumed per indicated horsepower per hour:
  (*a*) By engine and auxiliaries.....lb.
  (*b*) By main engine alone.....lb.
  (*c*) By auxiliaries.....lb.
(45) Dry steam consumed per brake horsepower per hour:
  (*a*) By engine and auxiliaries.....lb.
  (*b*) By main engine alone.....lb.
  (*c*) By auxiliaries.....lb.
(46) Percentage of steam used by main-engine cylinders accounted for by indicator diagrams:
  (*a*) Near cut-off.....per cent
  (*b*) Near release.....per cent

SAMPLE DIAGRAMS

TABLE 2 DATA AND RESULTS OF STEAM-ENGINE TEST COMPLETE FORM, CODE OF 1912

(1) Test of.....engine located at.....
  to determine.....conducted by.....
(2) Type of engine (simple, compound, or other multiple expansion; condensing or non-condensing).....
(3) Class of engine (mill, marine, electric, etc.).....
(4) Rated power of engine.....
(5) Name of builders.....
(6) Number and arrangement of cylinders of engine; how lagged; type of condenser.....
(7) Type of valves.....
(8) Type of boiler.....
(9) Kind and type of auxiliaries (air pump, circulating pump, feed pump; jackets, heaters, etc.).....

	1st Cyl.	2d Cyl.	3d Cyl.
(10) Dimensions of engine:			
(*a*) Single or double acting.....	.....	.....	.....
(*b*) Cylinder dimensions:			
Bore, in.....	.....	.....	.....
Stroke, ft.....	.....	.....	.....
Diameter of piston rod, in.....	.....	.....	.....
Diameter of tail rod, in.....	.....	.....	.....
(*c*) Clearance in per cent of volume displaced by piston per stroke:			
Head-end, per cent.....	.....	.....	.....

Crank-end, per cent.....
Average, per cent.....

(*d*) Surface in sq. ft. (average):
Barrel of cylinder, sq. ft.....
Cylinder heads, sq. ft.....
Clearance and ports, sq. ft.....
Ends of piston, sq. ft.....

(*e*) Jacket surfaces or internal surfaces of cylinder heated by jackets, in sq. ft.:
Barrel of cylinder, sq. ft.....
Cylinder heads, sq. ft.....
Clearance and ports, sq. ft.....
Receiver jackets, sq. ft.....

(*g*) Horsepower constant for 1 lb m.e.p. and 1 r.p.m.....

(11) Dimensions of boilers:
(*a*) Number .....
(*b*) Total grate surface.....sq. ft.
(*c*) Total water heating surface.....sq. ft.
(*d*) Total steam heating surface.....sq. ft.

(12) Dimensions of auxiliaries:
(*a*) Air pump.....
(*b*) Circulating pump.....
(*c*) Feed pumps.....
(*d*) Heaters .....

(13) Dimensions of condenser.....

(14) Dimensions of electric or other machinery driven by engine.....

(15) Date .....

(16) Duration .....hr.

AVERAGE PRESSURES AND TEMPERATURES

(17) Steam pressure at boiler by gage.....lb.

(18) Steam pipe pressure near throttle, by gage.....lb.

(19) Barometric pressure of atmosphere in lb. per. sq. in.....lb.

(20) Pressure in first receiver by gage.....lb.

(21) Pressure in second receiver by gage.....lb.

(22) Vacuum in condenser:
(*a*) In in. of mercury.....in.
(*b*) Corresponding total pressure.....lb.

(23) Pressure in steam jacket by gage.....lb.

(24) Pressure in reheater by gage.....lb.

(25) Superheat in steam leaving first receiver.....deg.

(26) Superheat in steam leaving second receiver.....deg.

(27) Temperature of main supply of feedwater to boilers.....deg.

(28) Temperature of additional supplies of feedwater.....deg.

(29) Ideal feedwater temperature corresponding to the pressure of the steam in the exhaust pipe, allowance being made for heat derived from jacket or reheater drips (British Standard)....deg.

(30) Temperature of injection or circulating water entering condenser.deg.
(31) Temperature of injection or circulating water leaving condenser.deg.
(32) Temperature of air in engine room.....deg.

TOTAL QUANTITIES

(33) Water fed to boilers from main source of supply.....lb.
(34) Water fed from additional supplies.....lb.
(35) Total water fed to boilers from all sources.....lb.
(36) Moisture in steam or superheating near throttle....per cent or deg.
(37) Factor of correction for quality of steam, dry steam being unity.lb.
(38) Total dry steam consumed for all purposes.....

HOURLY QUANTITIES

(39) Water fed from main source of supply.....lb.
(40) Water fed from additional supplies.....lb.
(41) Total water fed to boilers per hour.....lb.
(42) Total dry steam consumed per hour.....lb.
(43) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant.....lb.
(44) Net dry steam consumed per hour by engine and auxiliaries....lb.
(45) Dry steam consumed per hour:
(*a*) Main engine alone.....lb.
(*b*) Jackets and reheaters.....lb.
(*c*) Air pump.....lb.
(*d*) Circulating pump.....lb.
(*e*) Feedwater pump.....lb.
(*f*) Other auxiliaries.....lb.
(46) Injection or circulating water supplied condenser per hour....cu. ft.

HEAT DATA

(47) Heat units per pound of dry steam, based on temperature of Line 27.....B.t.u.
(48) Heat units per pound of dry steam, based on temperature of Line 28.....B.t.u.
(49) Heat units consumed per hour, main supply of feed.....B.t.u.
(50) Heat units consumed per hour, additional supplies of feed....B.t.u.
(51) Total heat units consumed per hour for all purposes.....B.t.u.
(52) Loss of heat per hour due to leakage of plant, drips, etc.....B.t.u.
(53) Net heat units consumed per hour:
(*a*) By engine and auxiliaries.....B.t.u.
(*b*) By engine alone.....B.t.u.
(*c*) By auxiliaries.....B.t.u.
(54) Heat units consumed per hour by the engine alone, reckoned from temperature given in Line 29 (British Standard)....B.t.u.

INDICATOR DIAGRAMS

		1st Cyl.	2d Cyl.	3d Cyl.
(55)	Commercial cut-off in per cent of stroke..	.....	.....	.....
(56)	Initial pressure in lb. per sq. in. above atmosphere	.....	.....	.....
(57)	Back-pressure at mid-stroke above or below atmosphere in lb. per sq. in.....	.....	.....	.....

(58) Mean effective pressure in lb. per sq. in... ....
(59) Equivalent mean effective pressure in lb. per sq. in.:
(*a*) Referred to first cylinder..... ....
(*b*) Referred to second cylinder..... ....
(*c*) Referred to third cylinder..... ....
(60) Pressures and percentages used in computing the steam accounted for by the indicator diagrams, measured to points on the expansion and compression curves:
Pressure above zero in lb. per sq. in:
(*a*) Near cut-off..... ....
(*b*) Near release..... ....
(*c*) Near beginning of compression ....
Percentage of stroke at points where pressures are measured:
(*d*) Near cut-off..... ....
(*e*) Near release..... ....
(*f*) Near beginning of compression ....
(61) Aggregate m.e.p. in lb. per sq. in. referred to each cylinder given in heading..... ....
(62) Mean back pressure above zero, lb. per sq. in..... ....
(63) Steam accounted for in lb. per indicated horsepower per hour:
(*a*) Near cut-off..... ....
(*b*) Near release..... ....
(64) Ratio of expansion..... ....
(65) Mean effective pressure of ideal diagram (Appendix No. 28)....lb.
(66) Diagram factor (Appendix No. 28).....

SPEED

(67) Revolutions per minute.....rev.
(68) Piston speed per minute.....ft.
(69) Variation of speed between no load and full load.....rev.
(70) Fluctuation of speed on suddenly changing from full load to no load, measured by the increase in the revolutions due to the change .....rev.

POWER

(71) Indicated horsepower developed by main engine:
1st cylinder.....i.h.p.
2d cylinder.....i.h.p.
3d cylinder.....i.h.p.
Whole engine.....i.h.p.
(72) Brake horsepower.....br.h.p.
(73) Friction i.h.p. by diagrams, no load on engine, computed for average speed.....i.h.p.
(74) Difference between Lines 71 and 72..... h.p.

(75) Percentage of i.h.p. of main engine lost in friction.....per cent
(76) Power developed by auxiliaries[1].....i.h.p.

ECONOMY RESULTS

(77) Heat units consumed per indicated horsepower per hour:[2]
  (*a*) By engine and auxiliaries.....B.t.u.
  (*b*) By engine alone.....B.t.u.
(78) Heat units consumed per brake horsepower per hour:
  (*a*) By engine and auxiliaries.....B.t.u.
  (*b*) By engine alone.....B.t.u.
(79) Heat units consumed by engine per hour, corresponding to ideal temperature of feedwater given in Line 29, per indicated horsepower (British Standard).....B.t.u.
(80) Dry steam consumed per i.h.p. per hour:
  (*a*) By engine and auxiliaries.....lb.
  (*b*) By main engine alone.....lb.
  (*c*) By auxiliaries.....lb.
(81) Dry steam consumed per brake h.p. per hour:
  (*a*) By engine and auxiliaries.....lb.
  (*b*) By main engine alone.....lb.
  (*c*) By auxiliaries.....lb.
(82) Percentage of steam used by main engine cylinders accounted for by indicator diagrams:

	1st Cyl.	2d Cyl.	3d Cyl.
(*a*) Near cut-off.....	.....	.....	.....
(*b*) Near release.....	.....	.....	.....

EFFICIENCY RESULTS

(83) Thermal efficiency ratio for engine and auxiliaries:
  (*a*) Per indicated horsepower.....per cent
  (*b*) Per brake horsepower.....per cent
(84) Thermal efficiency ratio for engine alone:
  (*a*) Per indicated horsepower.....per cent
  (*b*) Per brake horsepower.....per cent
(85) Ratio of economy of engine to that of an ideal engine working with the Rankine cycle (Appendix No. 31).....per cent

WORK DONE PER HEAT UNIT

(86) Ft.-lb. of net work per B.t.u. consumed by engine and auxiliaries (1,980,000 ÷ Line 78*a*).....ft.-lb.

[1] These are not included in the power developed by the main engine.

[2] The h.p. on which the economy and efficiency results are based are those of the main engine given in Line 71.

NOTE: Both the Short Form and Complete Form here given refer to a steam engine used for general service.

For an engine driving an electric generator the form should be enlarged to include the electrical data, embracing the average voltage, number of amperes each phase, number of watts, number of watt hours, average power factor, etc.; and the economy results based on the electrical output embracing the heat units and steam consumed per electric h.p. per hour and per kw-hr., together with the efficiency of the generator. See table for Steam Turbine Code, Part 6.

Likewise, in a marine engine having a shaft dynamometer, the form should include the data obtained from this instrument, in which case the brake h.p. becomes the shaft h.p.

PART VI

STEAM TURBINE CODE

RULES FOR CONDUCTING TESTS OF STEAM TURBINES AND TURBO-GENERATORS

1 OBJECT AND PREPARATIONS

Determine the object, take the dimensions, note the physical conditions not only of the turbine but of the entire plant concerned, examine for leakages, install the testing appliances, etc., as pointed out in the general instructions given on pages 1689 to 1714 and prepare for the test accordingly.

2 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple performance test of a steam turbine or turbo-generator, in which the steam consumption is determined by feedwater measurement, are:

(*a*) Tanks and platform scales for weighing water, (or water meters calibrated in place)
(*b*) Graduated scales attached to the water glasses of the boilers
(*c*) Pressure gages, vacuum gages, and thermometers
(*d*) A steam calorimeter
(*e*) A barometer
(*f*) A tachometer or other speed-measuring apparatus
(*g*) A friction brake or dynamometer
(*h*) Volt meters, ammeters, wattmeters, and watt-hour meters for the electrical measurements in the case of a turbo-generator

Directions regarding the use and calibration of these appliances are given on pages 1691 to 1702, and on page 1713.

The determination of the heat and steam consumption of a turbine or turbo-generator should conform to the same methods as those described in the Steam Engine Code, page 1735.

The steam consumed by steam-driven auxiliaries required for the operation of a turbine should be included in the total steam from which the heat consumption is calculated the same as in the case of the steam engine.

3 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view and see that they prevail throughout the trial.

4 DURATION

5 STARTING AND STOPPING

6 RECORDS

7 CALCULATION OF RESULTS

The rules pertaining to the subjects Duration, Starting and Stopping, Records, and Calculation of Results, are identically the same as those given under the respective headings in the Steam Engine Code, pages 1733 to 1737 with the single exception of the matter relating to indicator diagrams and results computed therefrom; and reference may be made to that code for the directions required in these particulars.

8 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view:

DATA AND RESULTS OF STEAM TURBINE OR TURBO-GENERATOR TESTS

CODE OF 1912

(1) Test of.....turbine located at.....
to determine....., conducted by.....
(2) Type of turbine and class of service.....
(3) Type of generator, kind of current, etc.....
(4) Rated power of turbine.....
(5) Type of boiler.....
(6) Kind and type of auxiliaries (air pumps, circulating pumps, feed pumps, etc.).....
(7) Dimensions of turbine or turbo-generator.....
(8) Dimensions of boilers .....
(9) Dimensions of auxiliaries .....
(10) Dimensions of condenser.....
(11) Date .....
(12) Duration .....hr.

AVERAGE PRESSURES AND TEMPERATURES

(13) Steam pipe pressure near throttle, by gage.....lb.

(14) Steam chest pressure by gage.....lb.
(15) Barometric pressure of atmosphere in lb. per sq. in.....lb.
(16) Vacuum in condenser:
(*a*) In inches of mercury.....in.
(*b*) Corresponding absolute pressure.....lb.
(17) Exhaust chamber pressure (absolute).....lb.
(18) Temperature of main supply of feedwater to boilers.....deg.
(19) Temperature of additional supplies of feedwater.....deg.
(20) Temperature of injection or circulating water entering condenser.....deg.
(21) Temperature of injection or circulating water leaving condenser.....deg.

TOTAL QUANTITIES

(22) Water fed to boilers from main source of supply.....lb.
(23) Water fed from additional supplies.....lb.
(24) Total water fed to boilers from all sources.....lb.
(25) Moisture in steam or superheating near throttle.....per cent or deg.
(26) Factor of correction for quality of steam, dry steam being unity.....
(27) Total dry steam consumed for all purposes.....lb.

HOURLY QUANTITIES.

(28) Water fed from main source of supply.....lb.
(29) Water fed from additional supplies.....lb.
(30) Total water fed to boilers per hour.....lb.
(31) Total dry steam consumed per hour.....lb.
(32) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant.....lb.
(33) Net dry steam consumed per hour.....lb.
(34) Dry steam consumed per hour:
(*a*) By turbine.....lb.
(*b*) By auxiliaries.....lb.
(35) Injection or circulating water supplied condensers per hour.....cu. ft.

HEAT DATA

(36) Heat units per pound of dry steam, based on temperature of Line 18.....B.t.u.
(37) Heat units per pound of dry steam, based on temperature of Line 19.....B.t.u.
(38) Heat units consumed per hour, main supply of feed.....B.t.u.
(39) Heat units consumed per hour, additional supplies of feed.....B.t.u.
(40) Total heat units consumed per hour for all purposes.....B.t.u.
(41) Loss of heat per hour due to leakage of plant, drips, etc.....B.t.u.
(42) Heat units consumed per hour:
(*a*) By turbine and auxiliaries.....B.t.u.
(*b*) By turbine alone.....B.t.u.
(*c*) By auxiliaries.....B.t.u.

Electrical Data

(43) Average volts, each phase.....volts
(44) Average amperes, each phase.....amperes
(45) Average kilowatts, first meter.....kw.
(46) Average kilowatts, second meter.....kw.
(47) Total kilowatt output.....kw.
(48) Power factor.....
(49) Output consumed by exciter.....kw.
(50) Net kilowatt output.....kw.

Speed

(51) Revolutions per minute.....rev.
(52) Variation of speed between no load and full load.....rev.
(53) Fluctuation of speed on suddenly changing from full load to no load, measured by the increase in the revolutions due to the change.....rev.

Power

(54) Brake horsepower.....br.h.p.
(55) Electrical horsepower.....h.p.

Economy Results

(56) Heat units consumed by turbine and auxiliaries per brake h.p.-hr.....B.t.u.
(57) Dry steam consumed per brake h.p.-hr.:
  (*a*) By turbine and auxiliaries.....lb.
  (*b*) By turbine alone.....lb.
  (*c*) By auxiliaries.....lb.
(58) Dry steam consumed per kw-hr.:
  (*a*) By turbine and auxiliaries.....lb.
  (*b*) By turbine alone.....lb.
  (*c*) By auxiliaries.....lb.

Efficiency Results

(59) Thermal efficiency ratio per brake horsepower.....per cent
(60) Ratio of economy of turbine to that of an ideal turbine working with the Rankine cycle.....

Work Done per Heat Unit

(61) Ft.-lb. of net work per B.t.u. consumed by turbine and auxiliaries (1,980,000 ÷ Line 56).....ft-lb.

PART VII

CODE FOR COMPLETE STEAM POWER PLANTS

RULES FOR CONDUCTING TESTS OF COMPLETE STEAM POWER PLANTS

1 OBJECT AND PREPARATIONS

These rules are intended to apply to commercial tests of a complete plant to determine the number of pounds of fuel consumed per unit of work done in a unit of time. For tests of the component parts of a complete plant, such as boilers, engines, turbines, etc., rules may be found in the respective Codes applying to such cases.

Read the general instructions given on pages 1689 to 1704. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

2 FUEL

Determine the character of the fuel to be used according to the object in view. For further particulars reference may be made to the Boiler Code, page 1715.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple performance test of a steam plant are:

(*a*) Platform scales for weighing coal and ashes
(*b*) Coal calorimeter
(*c*) Steam engine indicators
(*d*) A tachometer or other speed-measuring apparatus
(*e*) Electrical instruments for determining the output of an electric plant

Directions regarding the use and calibration of these appliances are given on pages 1691 to 1702.

If the test involves the determination of boiler performance, and engine or turbine performances, additional instruments should be used as pointed out in the respective Codes referring to such tests.

4 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to

the object in view, and see that they prevail throughout the trial, as pointed out on page 1703.

5 DURATION

The duration of a plant test should be not less than one day of 24 hours, and preferably a full week of seven days, including Sunday.

In cases where the engine or turbine is in operation only a part of the day, the duration on which the results are computed, should be considered the length of time that the engine or turbine is in operation at its working speed.

6 STARTING AND STOPPING

In a plant operating continuously, day and night, the times fixed for starting and stopping should follow the regular periods of cleaning the fires. The fires should be quickly cleaned and then burned low, say to a thickness of 4 in. When this condition is reached the time should be noted as the starting time, and the thickness of each coal bed observed, as also the water levels and the steam pressure. Fresh coal should then be fired from that weighed for the test, the ashpits thoroughly cleaned, and the regular work of the test proceeded with. At the close of the test, following a regular cleaning, the fires should again be burned low, and when their condition has become the same as that observed at the beginning, the water levels and steam pressure also being the same, the time is observed and this time taken as the stopping time. If the water levels and steam pressure are not the same as at the beginning a suitable correction should be made by computation. The ashes and refuse are then hauled from the ashpits.

In a plant running only a part of the day, and during the balance of the day the fires are banked, the time selected for the beginning and end of the test should be that following the close of the day's run, when the fires have been burned low preparatory to cleaning and banking. The amount of live coal left on the grates under these circumstances is estimated at the beginning of the test, and the fires brought to the same condition, as near as may be, at the close of the test the next day. If the two quantities differ, a suitable correction is made in the weight of coal fired, as found by calculation.

7 RECORDS

The general data should be recorded as pointed out in Part I, page 1703, under the head of Records. Half-hourly readings of the

various instruments concerned are usually sufficient, excepting where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes it necessary. Mark on each card the cylinder and the end on which it was taken, also the time of the day. Record on one card of each set the readings of the pressure gages concerned, taken at the same time. These records should subsequently be entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are the worked up.

Further directions may be found in Part I, page 1699, under the heading Indicators and in Appendix No. 18.

8 SAMPLING AND DRYING COAL

During the progress of the test the coal should be regularly sampled for the purpose of analysis and determination of moisture, in the manner pointed out in Part III, page 1712.

9 ASHES AND REFUSE

The ashes and refuse withdrawn from the furnace and ashpit during the progress of the test and at its close should be weighed in a dry state, and, if desired, a representative sample should be obtained for proximate analysis and the determination of the amount of unburned carbon which it contains.

10 CALORIFIC TESTS AND ANALYSES OF COAL

The quality of the fuel should be determined by calorific tests and analysis of the representative sample above referred to. Directions for making these tests and analyses will be found in Part I under the headings Coal Calorimeters and Coal Analysis Apparatus, page 1697.

11 CALCULATION OF RESULTS

The methods of calculating the indicated and electrical horsepowers are the same as explained in the Steam Engine Code, page 1736.

12 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

DATA AND RESULTS OF COMPLETE STEAM POWER PLANT TEST

(1) Test of..... plant located at.....
to determine....., conducted by.....
(2) Type of engine or turbine and class of service.....
(3) Rated power of engine or turbine.....
(4) Type of boilers.....
(5) Kind and type of auxiliaries (air pump, circulating pump, and feed pump; jackets, heaters, etc.).....
(6) Dimensions of engine or turbine.....
(7) Dimensions of boilers .....
(8) Dimensions of auxiliaries .....
(9) Dimensions of condenser .....
(10) Date .....
(11) Duration .....hr.
(12) Length of time engine or turbine was in motion with throttle open .....hr.
(13) Length of time engine or turbine was running at normal speed...hr.
(14) Kind of coal.....
(15) Size of coal.....

Average Pressures and Temperatures

(16) Steam pressure at boiler by gage.....lb.
(17) Steam pipe pressure near throttle, by gage.....lb.
(18) Barometric pressure of atmosphere in in. of mercury.....in.
(19) Pressure in receiver by gage.....lb.
(20) Vacuum in condenser.....in.
(21) Number of degrees of superheating, if any, near throttle.....deg.
(22) Temperature of feedwater entering boilers.....deg.

Total Quantities, Time, Etc.

(23) Total coal as fired[1].....lb.
(24) Moisture in coal.....per cent
(25) Total dry coal consumed.....lb.
(26) Ash and refuse.....lb.
(27) Percentage of ash and refuse to dry coal.....per cent
(28) Calorific value by calorimeter test per lb. of dry coal.....B.t.u.
(29) Cost of coal per ton of .... lb.....dollars

Hourly Quantities

(30) Dry coal consumed per hour, based on duration of running period .....lb.

Indicator Diagrams

(31) Mean effective pressure in lb. per sq. in.....lb.

Electrical Data

(32) Total electrical output.....kw-hr.
(33) Electrical output per hour.....kw.
(34) Output consumed by exciter.....kw.

[1] Where an independent superheater is used, this includes coal burned in the superheater.

(35) Net electrical output per hour.....kw.
(36) Average volts each phase.....volts
(37) Average amperes each phase.....amperes
(38) Power factor.....

SPEED

(39) Revolutions per minute.....rev.

POWER

(40) Indicated horsepower developed by main engine:
First cylinder.....i.h.p.
Second cylinder.....i.h.p.
Whole engine.....i.h.p.
(41) Net electrical horsepower.....h.p.

ECONOMY RESULTS

(42) Dry coal consumed per i.h.p. per hour .....lb.
(43) Dry coal consumed per kw-hr. .....lb.
(44) Cost of coal per i.h.p. per hour.....cents
(45) Cost of coal per kw-hr. .....cents

PART VIII

PUMPING MACHINERY CODE

RULES FOR CONDUCTING DUTY TRIALS OF STEAM PUMPING MACHINERY [1]

1 OBJECT AND PREPARATIONS

Read the general instructions given on pages 1689 to 1714. Determine the object, take the dimensions, note the physical conditions not only of the pumping machinery but of all parts of the plant concerned, examine for leakages, install the testing appliances, etc., as there pointed out, and prepare for the test accordingly.

In a reciprocating pump, determine the quantity of water leakage or slip past the plungers, and that of the pump valves, if any, as pointed out in Appendix No. 32.

2 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple duty trial of pumping machinery, in which the steam consumption is determined by feedwater measurement, are:

(*a*) Tanks and platform scales for weighing water (or water meters calibrated in place)
(*b*) Graduated scales attached to the water glasses of the boilers
(*c*) Pressure gages, vacuum gages, and thermometers
(*d*) A steam calorimeter
(*e*) A barometer
(*f*) A tachometer or other speed-measuring apparatus
(*g*) For rotary pumps a weir or other means for measuring the quantity of water pumped
(*h*) Stroke scales, for direct-acting pumps

In trials of a reciprocating pumping engine, involving the determination of the complete performance, the weir or other means of measurement noted should be provided, and in addition the following:

(*i*) Steam engine indicators
(*j*) A planimeter

[1] In the case of a pump driven by some other prime mover than a steam engine or steam turbine, the code may be modified to suit the particular circumstances.

Directions regarding the use and calibration of these appliances are given on pages 1691 to 1702 and page 1713.

The determination of the heat and steam consumption should conform to the same methods as those described in the Steam Engine Code, pages 1732 and 1733.

The steam consumed by steam-driven auxiliaries which are required in the operation of the pumping machinery should be included in the total steam from which the heat consumption is calculated, the same as noted in the Steam Engine Code.

3 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view and see that they prevail throughout the trial, as pointed out on page 1703.

In trials for maximum duty, care should be taken that no air is snifted into the pump cylinders, causing imperfect filling. In such cases, and indeed in all cases where air is thus admitted in sufficient quantity to affect the performance as revealed by indicator diagrams from the water end, the result should be corrected accordingly.

4 DURATION

5 STARTING AND STOPPING

6 RECORDS

The rules pertaining to the subjects Duration, Starting and Stopping, and Records, are identically the same as those given under the respective headings in the Steam Engine Code, pages 46, 47 and 48; and reference may be made to that code for the necessary directions in these particulars. Where the pump-end is of the reciprocating class, the indicator diagrams should be taken not only from the steam cylinders but also from the water cylinders.

7 CALCULATION OF RESULTS

The rules pertaining to Dry Steam, Heat Consumption, and Indicated Horsepower, are identically the same as those given on pages 1735 and 1736 of the Steam Engine Code; and reference may be made to that code for the necessary directions in these particulars.

(*a*) *Water Horsepower.* The water horsepower in a reciprocating pump is found by multiplying the net area of the plunger in sq. in.

by the total head, which is made up of the pressure shown by the gage on the force main, that on the suction main, and that representing the vertical distance between the centers of the two gages, all expressed in lb. per sq. in.; the length of the stroke in ft.; and the number of single strokes per minute; and dividing the final product by 33,000.

In a rotary pump the water horsepower is found by multiplying the weight of water discharged per hour in lb., as determined by weir or other measurement; by the total head in ft., as determined from the readings of the gage on the force main, that on the suction main, and the vertical distance between the two gages; and the product divided by 1,980,000.[1]

(*b*) *Duty.* The duty per million heat units is found by dividing the number of ft-lb. of work done during the trial by the total number of heat units consumed; and multiplying the quotient by 1,000,000. The amount of work is found in the case of reciprocating pumps by multiplying the net area of the plunger in sq. in., the total head expressed in lb. per sq. in. (which is made up of the pressure shown by the gage on the force main, that on the suction main, and the vertical distance between the centers of the two gages, all reduced to lb.), by the length of the stroke in ft., and the total number of single strokes during the trial; finally correcting for the percentage of leakage of the pump. In a rotary pump the work done is found by multiplying the weight of water discharged during the trial, as determined by weir or other measurement, by the total head in ft.

The duty per 1000 lb. of dry steam is found by dividing the ft-lb. of work done, as noted above, by the total weight of dry steam, and multiplying the quotient by 1000.

(*c*) *Capacity.* The capacity in gal. per 24 hr. for reciprocating pumps is found by multiplying the net area of the plunger by the length of the stroke in ft., (in direct-connected engines the average length of stroke); then by the number of single strokes per minute; and the product of these three by the constant 74.8; finally correcting for the percentage of leakage of the pump.

(*d*) *Leakage of Pump.* The percentage of leakage is the percentage borne by the quantity of leakage found on the leakage trial, to the quantity of water discharged on the duty run determined from plunger displacement.

(*e*) *Friction.* The percentage of total friction in a reciprocating pump is the percentage borne by the friction horsepower to the indicated horsepower of the steam cylinders.

(*f*) *Miscellaneous.* For the calculation of other results pertaining specially to the performance of the steam end of a reciprocating pump, reference may be made to the Steam Engine Code.

[1] If there is a material difference in velocity of the water at the points where the gages are attached, a correction should be made for the corresponding difference in "velocity head."

8 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

In the case of a pumping engine of the reciprocating class for which a record of the complete performance is desired, the additional engine data and results given in the Complete Form of the Steam Engine Code may supplement those here given.

DATA AND RESULTS OF DUTY TRIAL OF STEAM PUMPING MACHINERY

CODE OF 1912

(1) Test of.....pump, located at.....
to determine....., conducted by.....
(2) Type of machinery.....
(3) Rated capacity in gal. per 24 hr.....
(4) Type of boiler.....
(5) Type of auxiliaries.....
(6) Dimensions of engine or turbine.....
(7) Dimensions of pump.....
(8) Dimensions of boilers.....
(9) Dimensions of auxiliaries.....
(10) Dimensions of condenser.....
(11) Date .....
(12) Duration .....hr.

AVERAGE PRESSURES AND TEPERATURES

(13) Steam pressure at boiler by gage.....lb. per sq. in.
(14) Steam pipe pressure near throttle, by gage.....lb. per sq. in.
(15) Barometric pressure of atmosphere in in. of mercury.....in.
(16) Pressure in receiver by gage.....lb. per sq. in.
(17) Vacuum in condenser in in. of mercury.....in.
(18) Pressure in force main by gage.....lb.
(19) Pressure in suction main by gage.....lb.
(20) Vertical distance between centers of two gages.....ft.
(21) Total head, expressed in lb. pressure.....lb.
(22) Total head, expressed in ft.....ft.
(23) Temperature of main supply of feedwater to boilers.....deg.
(24) Temperature of additional supplies of feedwater.....deg.
(25) Temperature of air in engine room.....deg.

TOTAL QUANTITIES

(26) Water fed to boilers from main source of supply.....lb.
(27) Water fed from additional supplies.....lb.
(28) Total water fed to boilers from all sources.....lb.

(29) Moisture in steam or superheating near throttle.....per cent or deg.
(30) Factor of correction for quality of steam, dry steam being unity.....
(31) Total dry steam consumed for all purposes.....lb.
(32) Total leakage of plungers and valves.....gal.
(33) Total number of gallons of water discharged:
(*a*) By plunger displacement, uncorrected.....gal.
(*b*) By plunger displacement, corrected for leakage.....gal.
(*c*) By weir or other measurement of discharge.....gal.
(34) Total weight of water discharged:
(*a*) By plunger displacement, corrected.....lb.
(*b*) By measurement of discharge.....lb.

HOURLY QUANTITIES

(35) Water fed from main source of supply.....lb.
(36) Water fed from additional supplies.....lb.
(37) Total water fed to boilers per hour.....lb.
(38) Total dry steam consumed per hour.....lb.
(39) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant.....lb.
(40) Net dry steam consumed per hour.....lb.
(41) Dry steam consumed per hour:
(*a*) By main engine or turbine.....lb.
(*b*) By auxiliaries.....lb.
(42) Water discharged per hour:
(*a*) By plunger displacement, corrected.....lb.
(*b*) By measurement of discharge.....lb.

HEAT DATA

(43) Heat units per lb. of dry steam based on temperature, Line 23..B.t.u.
(44) Heat units per lb. of dry steam based on temperature, Line 24..B.t.u.
(45) Heat units consumed per hour based on main supply of feed..B.t.u.
(46) Heat units consumed per hour based on additional supplies of feed .....B.t.u.
(47) Total heat units consumed per hour for all purposes.....B.t.u.
(48) Loss of heat per hour due to leakage of plant, drips, etc.....B.t.u.
(49) Net heat units consumed per hour.....B.t.u.
(50) Heat units consumed per hour:
(*a*) By engine or turbine alone.....B.t.u.
(*b*) By auxiliaries.....B.t.u.

INDICATOR DIAGRAMS

(51) Mean effective pressure.....lb.

SPEED AND STROKE

(52) Revolutions per minute.....rev.
(53) Number of single strokes per minute.....strokes
(54) Average length of stroke.....ft.

POWER

(55) Indicated horsepower developed:
1st cylinder.....i.h.p.
2d cylinder.....i.h.p.
Whole engine.....i.h.p.

(56) **Water horsepower**.....h.p.
(57) **Friction h.p. (Line 55—Line 56)**.....fr. h.p.
(58) **Percentage of i.h.p. lost in friction**.....per cent

DUTY

(59) **Duty per million heat units**.....ft.-lb.
(60) **Duty per thousand lb. of steam**.....ft.-lb.

WORK DONE PER HEAT UNIT

(61) Ft.-lb. of work per B.t.u. (Line 59 ÷ 1,000,000).....ft.-lb.

CAPACITY

(62) Number of gal. of water pumped in 24 hr.:
  (*a*) By plunger displacement, corrected for leakage.....gal.
  (*b*) By weir or other measurement.....gal.
(63) Gallons of water per minute:
  (*a*) By plunger displacement, corrected for leakage.....gal.
  (*b*) By weir or other measurement.....gal.

ECONOMY RESULTS, STEAM END

(64) Heat units consumed per i.h.p. per hour:[1]
  (*a*) By engine and auxiliaries.....B.t.u.
  (*b*) By engine alone.....B.t.u.
  (*c*) By auxiliaries.....B.t.u.
(65) Dry steam consumed per i.h.p. per hour:
  (*a*) By engine and auxiliaries.....lb.
  (*b*) By engine alone.....lb.
  (*c*) By auxiliaries.....lb.

[1] The i.h.p. on which the economy results are based is that of the main engine given in Line 54.

PART IX

CODE FOR COMPLETE PUMPING MACHINERY PLANTS

RULES FOR CONDUCTING TESTS OF COMPLETE STEAM PUMPING MACHINERY PLANTS

1 OBJECT AND PREPARATIONS

This code applies to a commercial test of a complete steam pumping machinery plant, having for an object the determination of the fuel cost of pumping a given quantity of water for a day's run of 24 hours. For tests of the component parts of the plant, rules may be found in the respective codes applying thereto.

Read the general instructions given on pages 1689 to 1704. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out and prepare for the test accordingly.

In a reciprocating pump, ascertain the quantity of water leakage or slip past the plungers, and that of the pump valves, if any, as pointed out in Appendix No. 32.

2 FUEL

Determine the character of the fuel to conform with the object in view. To obtain maximum economy or capacity, the fuel should be some kind of coal that is regarded as a standard, as noted in the Boiler Code, page 1715.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple performance test of a complete pumping machinery plant are:

(*a*) Platform scales for weighing coal and ashes
(*b*) A coal calorimeter
(*c*) A tachometer or other speed measuring apparatus.
(*d*) A pressure gage attached to the force main
(*e*) For rotary pumps, a weir or other means for measuring the quantity of water pumped
(*f*) Stroke scales, for direct-acting pumps.

Directions regarding the use and calibration of these appliances may be found on pages 1691 to 1702.

If the test also involves the determination of boiler performance and pumping machinery performance, additional apparatus and instruments are required, as pointed out in the respective codes referring thereto.

4 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial, as explained on page 1703, Part I.

5 DURATION

The duration of a test of a complete pumping machinery plant should be not less than one day of 24 hours.

In cases where the machinery is in operation only a part of the calendar day, the coal consumption used in computing the results should be that of the entire 24 hours.

6 MISCELLANEOUS

The methods of starting and stopping a test of a complete pumping plant, sampling and drying coal, determining ashes and refuse, and ascertaining the calorific value and chemical analysis of the coal, are the same as those described in the Code for Complete Steam Power Plants, pages 1753 and 1754.

7 CALCULATION OF RESULTS

For the calculation of water horsepower, duty, capacity, and leakage of pump, reference may be made to the explanations given on pages 1759 and 1760 of the Pumping Engine Code.

8 RECORDS, DATA, AND RESULTS

The records should be made as explained on page 1703, and the tabular summary of data and results reported in accordance with the form given herewith, adding lines not provided for, or omitting those not required.

DATA AND RESULTS OF TEST OF STEAM PUMPING MACHINERY PLANT

CODE OF 1912

(1) Test of.....plant located at.....
to determine.....conducted by.....
(2) Type of machinery.....
(3) Rated capacity in gallons per 24 hrs.....
(4) Type of boilers .....
(5) Type of auxiliaries .....
(6) Dimensions of machinery .....
(7) Dimensions of boilers .....
(8) Dimensions of auxiliaries .....
(9) Dimensions of condenser .....
(10) Date .....
(11) Duration .....hr.
(12) Length of time machinery was in motion with throttle open....hr.
(13) Length of time machinery was running at normal speed.....hr.

Average Pressures and Temperatures

(14) Steam pressure at boiler by gage.....lb. per sq. in.
(15) Pressure in force main by gage.....lb.
(16) Pressure in suction main by gage.....in.
(17) Vertical distance between centers of two gages.....ft.
(18) Total head expressed in lb. pressure.....lb.
(19) Total head expressed in ft.....ft.

Speed and Stroke

(20) Revolutions per minute.....rev.
(21) Number of single strokes per minute.....strokes
(22) Average length of stroke.....ft.

Total Quantities.

(23) Total coal as fired.....lb.
(24) Moisture in coal.....per cent
(25) Total dry coal consumed.....lb.
(26) Ash and refuse.....lb.
(27) Percentage of ash and refuse to dry coal.....per cent
(28) Calorific value per lb. of dry coal by calorimeter test.....B.t.u.
(29) Cost of coal per ton of ..... lb.....dollars
(30) Total leakage of plungers and valves.....gal.
(31) Total number of gals of water discharged:
(*a*) By plunger displacement, uncorrected.....gal.
(*b*) By plunger displacement, corrected for leakage.....gal.
(*c*) By weir or other measurement.....gal.

Hourly Quantities

(32) Dry coal consumed per hour, based on duration of running period, (Line 25÷Line 12).....lb.

POWER

(33) Water horsepower.....h.p.

ECONOMY RESULTS

(34) Dry coal consumed per water h.p. per hour.....lb.
(35) Cost of coal per water h.p. per hour.....cents
(36) Duty per 100 lb. of dry coal.....

CAPACITY

(37) Number of gal. of water pumped in 24 hr.:
  (*a*) By plunger displacement, corrected for leakage.....gal.
  (*b*) By weir or other measurement.....gal.
(38) Number of gal. of water pumped per minute:
  (*a*) By plunger displacement, corrected for leakage.....gal.
  (b) By weir or other measurement.....gal.

PART X

CODE FOR AIR MACHINERY

RULES FOR CONDUCTING TESTS OF STEAM-DRIVEN COMPRESSORS, BLOWERS AND FANS[1]

1 OBJECT AND PREPARATIONS

2 APPARATUS AND INSTRUMENTS

3 OPERATING CONDITIONS

4 DURATION

5 STARTING AND STOPPING

6 RECORDS

The directions pertaining to the above divisions of the subject are almost identical with those given in the Pumping Machinery Code under the same headings, the only difference being that the work done by the steam end is expended in moving a volume of air, instead of a body of water.

The quantity of air discharged should be measured by a gasometer, or by delivery into tanks of known capacity. Where these means are not available the pitot tube should be used, as described in Appendix No. 6, or some other means should be employed which is subject to calibration.

If the air end is of the reciprocating type, indicator diagrams should be regularly taken from this end as well as from the steam end.

7 CALCULATION OF RESULTS

The rules pertaining to dry steam, heat consumption, and indicated horsepower of the steam end, are identically the same as those given on pages 1735 and 1736 of the Steam Engine Code, and reference may be made to that code for the necessary directions in these particulars.

(*a*) *Air Horsepower.* The gross work done at the air end of a reciprocating machine, expressed in horsepower, is found by multiplying together the net area of the air piston in sq. in., the mean effective air pressure in lb. per sq. in. as determined from indicator diagrams, the

[1] In the case of air machinery driven by some other prime mover than a steam engine or turbine, the code may be modified to meet the particular requirements.

length of the stroke in ft., and the number of single strokes per minute; and dividing their product by 33,000.

The net work at the air end of either reciprocating or rotary machines, expressed in ft-lb. per minute, is found by multiplying the corrected volume of the compressed air in cu. ft. discharged into the main delivery pipe per minute, by the impact or total pressure in lb. per sq. ft. and by the hyperbolic logarithm of the ratio of the total pressure to the atmospheric pressure (all pressures being absolute pressures). The net air horsepower is found by dividing the product by 33,000. The corrected volume of the compressed air may be found by multiplying the sectional area of the delivery main in sq. ft. by the mean velocity in ft. per minute as determined by pitot tube or other measurement, and reducing the result to atmospheric temperature by multiplying by the proportion,

$$\frac{460 + t}{460 + T}$$

in which t is the temperature of the air supplied to the machine and T the temperature of the air in the delivery main.

(*b*) *Capacity.* The capacity is the number of cu. ft. of air discharged through the delivery main per minute, as determined by gasometer, tank or other mode of measurement, reduced to the equivalent free air at the atmospheric temperature and pressure. The correction for pressure is made by multiplying by the proportion $\frac{P_2}{P_1}$, in which P_1 is the atmospheric pressure and P_2 the total pressure in the main (absolute pressures), and the correction for temperature as above.

The capacity may also be expressed in the number of cu. ft. of compressed air discharged per minute at a given pressure above the atmosphere reduced to the atmospheric temperature.

(*c*) *Miscellaneous.* For methods of calculating results pertaining especially to the performance of the steam-end of a reciprocating air pumping machine, reference may be made to the Steam Engine Code, pages 1735, 1736, and 1737.

The "efficiency of compression" in a reciprocating machine is determined by first ascertaining the net work at the air end given above under the heading, "(*a*) Air Horsepower," and then dividing the net work thus found by the gross work given under the same heading.

The "mechanical efficiency" of a reciprocating machine is determined by dividing the gross air horsepower at the air end by the indicated horsepower at the steam end, or by the horsepower delivered by the belt or motor in the case of other means of driving.

8 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for and omitting those not required, as may conform with the object in view.

In the case of an air-pumping machine of the reciprocating class for which a record of the complete performance is desired, the additional engine data and results given in the Complete Form of the Steam Engine Code may supplement those here given:

DATA AND RESULTS OF TEST OF AIR MACHINERY

CODE OF 1912

(1) Test of.....located at.....
to determine.....conducted by.....
(2) Type of machinery.....
(3) Rated capacity in cu. ft. of free air per minute.....
(4) Rated capacity in cu. ft. of air discharged per minute at 100 lb. per sq. in. above atmosphere, reduced to the atmospheric temperature .....cu. ft.
(5) Type of boilers .....
(6) Type of auxiliaries .....
(7) Dimensions of engine or turbine at steam end.....
(8) Dimensions of cylinders or blowers at air end.....
(9) Dimensions of boilers .....
(10) Dimensions of auxiliaries .....
(11) Dimensions of condenser .....
(12) Date .....
(13) Duration .....hr.

Average Pressures and Temperatures

(14) Steam pressure at boiler by gage.....lb. per sq. in.
(15) Steam pipe pressure near throttle, by gage.....lb. per sq. in.
(16) Barometric pressure of atmosphere in in. of mercury.....in.
(17) Pressure in receiver by gage.....lb. per sq. in.
(18) Vacuum in condenser in in. of mercury.....in.
(19) Pressure in delivery main by gage (impact pressure).....lb.
(20) Total head, expressed in ft.....ft.
(21) Temperature of main supply of feedwater to boilers.....deg.
(22) Temperature of additional supplies of feedwater.....deg.
(23) Temperature of air in engine room or air supplied to machine..deg.
(24) Temperature by wet-bulb thermometer.....deg.
(25) Temperature of air in delivery main.....deg.

Total Quantities

(26) Water fed to boilers from main source of supply.....lb.
(27) Water fed from additional supplies.....lb.
(28) Total water fed to boilers from all sources.....lb.
(29) Moisture in steam or superheating near throttle....per cent or deg.
(30) Factor of correction for quality of steam, dry steam being unity...
(31) Total dry steam consumed for all purposes.....lb.
(32) Total cu. ft. of compressed air delivered as measured.....cu. ft.

(33) Total cu. ft. of compressed air delivered reduced to atmospheric temperature and pressure.....lb.
(34) Total weight of air delivered.....lb.

HOURLY QUANTITIES

(35) Water fed from main source of supply.....lb.
(36) Water fed from additional supplies.....lb.
(37) Total water fed to boilers per hour.....lb.
(38) Total dry steam consumed per hour.....lb.
(39) Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant.....lb.
(40) Net dry steam consumed per hour.....lb.
(41) Dry steam consumed per hour:
(*a*) By main engine or turbine.....lb.
(*b*) By auxiliaries.....lb.
(42) Cu. ft. of compressed air delivered per hour as measured.....cu. ft.
(43) Cu. ft. of compressed air delivered per hour reduced to atmospheric temperature.....cu. ft.
(44) Cu. ft. of compressed air delivered per hour reduced to atmospheric temperature and pressure.....cu. ft.
(45) Weight of air delivered per hour.....lb.

HEAT DATA

(46) Heat units per lb. of dry steam based on temperature Line 21..B.t.u.
(47) Heat units per lb. of dry steam based on temperature Line 22..B.t.u.
(48) Heat units consumed per hour based on main supply of feed...B.t.u.
(49) Heat units consumed per hour based on additional supplies of feed.....B.t.u.
(50) Total heat units consumed per hour for all purposes.....B.t.u.
(51) Loss of heat per hour due to leakage of plant, drips, etc....B.t.u.
(52) Net heat units consumed per hour.....B.t.u.
(53) Heat units consumed per hour:
(*a*) By engine or turbine alone.....B.t.u.
(*b*) By auxiliaries.....B.t.u.

INDICATOR DIAGRAMS

(54) Mean effective pressure in steam cylinders.....lb.
(55) Mean effective pressure in air cylinders.....lb.

SPEED AND STROKE

(56) Revolutions per minute.....rev.
(57) Number of single strokes per minute.....strokes

POWER

(58) Indicated horsepower developed at steam end of reciprocating machine.....i.h.p.
(59) Gross air horsepower as indicated in air cylinders of reciprocating machine.....air h.p.
(60) Net air horsepower as computed from Line 43.....air h.p.
(61) Friction of reciprocating machine (Line 58 — Line 59)....fr. h.p.
(62) Percentage of i.h.p. lost in friction of machine.....per cent

Economy Results, Steam End of Engine-Driven Machines

(63) Heat units consumed per i.h.p. per hour:
- (*a*) By engine and auxiliaries.....B.t.u.
- (*b*) By engine alone.....B.t.u.
- (*c*) By auxiliaries.....B.t.u.

(64) Dry steam consumed per i.h.p. per hour[1]:
- (*a*) By engine and auxiliaries.....lb.
- (*b*) By engine alone.....lb.
- (*c*) By auxiliaries.....lb.

Economy Results, Air Delivered

(65) Heat units consumed per hour per net air h.p. of Line 60:
- (*a*) By engine or turbine and auxiliaries.....B.t.u.
- (*b*) By engine or turbine alone.....B.t.u.
- (*c*) By auxiliaries.....B.t.u.

(66) Dry steam consumed per hour per net h.p. of Line 60:
- (*a*) By engine or turbine and auxiliaries.....lb.
- (*b*) By engine or turbine alone.....lb.
- (*c*) By auxiliaries.....lb.

Efficiency Results

(67) Thermal efficiency ratio for engine alone:
- (*a*) Per i.h.p., steam end (2545 ÷ Line 63*b*).....per cent
- (*b*) Per net air h.p., air delivery (2545 ÷ Line 65*b*)....per cent

Work Done per Heat Unit

(68) Ft.-lb. of net work per B.t.u. consumed by engine or turbine and auxiliaries (1,980,000 ÷ Line 65*a*).....ft-lb.

Capacity

(69) Cu. ft. of compressed air delivered per minute as measured....cu. ft.

(70) Cu. ft. of compressed air delivered per minute, reduced to atmospheric temperature.....cu. ft.

(71) Cu. ft. of compressed air delivered per minute at 100 lb. pressure, reduced to atmospheric temperature.....cu. ft.

(72) Cu. ft. of compressed air delivered per minute reduced to atmospheric temperature and pressure (free air).....cu. ft.

Miscellaneous Results

Steam Driven Reciprocating Machine

(73) Efficiency of compression (Line 60 ÷ Line 59 × 100).....per cent

(74) Mechanical efficiency of machine (Line 59 ÷ Line 58 × 100).per cent

(75) Volumetric efficiency (Line 72 ÷ 1st Compr. Displ. × 100)...per cent

[1] The i.h.p. on which these economy results are based is that of the main engine given in Line 58.

Note: In the case of air compressors having more than one stage and in those having intercoolers, additional dat should be given covering pressures and temperatures in the different stages, the quantity of water used for cooling and temperatures of the air and water entering and leaving the cooler.

PART XI

LOCOMOTIVE CODES

RULES FOR CONDUCTING TESTS OF LOCOMOTIVES

CODE FOR SHOP TESTS

1 INTRODUCTION

Locomotive tests are of two leading kinds, shop tests and road tests. The former are made under conditions quite similar to those of a stationary testing plant in which the power is absorbed by a brake. The latter are made under conditions of service on the road, the locomotive hauling a train of cars. Being a complete steam plant in itself, embracing boiler, engine, and certain auxiliaries, a locomotive equipped for a shop test should be considered in the same class as a stationary plant, and the code of rules for such tests are therefore similar to those given in Part VII.

2 OBJECT AND PREPARATIONS

The leading object of a shop test, so far as covered by this code, is the determination of the coal and steam consumption per unit of power when the locomotive is operated under fixed conditions. There are other less prominent objects, such as a determination of the performance of the engine alone under different conditions, or of the boiler running under various rates of combustion, but tests having these objects are substantially covered by the Boiler and Engine Codes of Parts IV and V, to which reference may be made.

Note the general instructions given on pages 1689 to 1714 so far as they pertain to the work in hand. Take the dimensions, note the physical conditions, examine for leakages, install the testing appliances, etc., as there pointed out, and make preparations for the test accordingly.

It is assumed that a testing plant is available where the driving wheels can be mounted upon friction brake apparatus for suitably disposing of and measuring the power given off, as further explained in Appendix No. 33.

3 FUEL[1]

Select the fuel required in accordance with the special object in view. If maximum efficiency or capacity is desired the fuel should be some kind of coal that is regarded as a standard for the locality where the test is made, as noted in the Boiler Code, page 1715.

4 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a shop test of a locomotive are

(*a*) Platform scales for weighing coal and ashes
(*b*) A coal calorimeter
(*c*) Tanks and scales for weighing water (or water meters calibrated in place)
(*d*) A graduated scale attached to the water glass
(*e*) Pressure gages, draft gages, and thermometers
(*f*) A steam calorimeter
(*g*) Steam engine indicators
(*h*) A planimeter
(*i*) A tachometer or other speed-measuring apparatus
(*j*) Friction brake apparatus for absorbing and measuring the power given off
(*k*) A dynamometer for determining the pull on the drawbar

Directions regarding their use and calibration are given on pages 1691 to 1702, page 1713, and Appendix No. 34.

5 OPERATING CONDITIONS

Determine what the operating conditions and method of firing should be as explained on page 1703 and see that they prevail throughout the trial.

6 DURATION

The duration of a simple shop test of a locomotive depends somewhat upon the rate of combustion, but it should not be less than two hours. In any case it should continue until at least 250 lb. of coal have been burned on each square foot of grate.

On account of liability of error in estimating the difference in the amount of coal in the furnace at beginning and end, the results of a single test should not be relied upon, but repeated tests should be made under the required conditions and the results averaged.

[1] See Section 14, Page 1781, for memoranda relating to oil fuel.

7 STARTING AND STOPPING

Set the engine to work with a comparatively light load at the normal speed. Burn the fire down as low as practicable and clean it of ashes and clinkers by rocking the grate, afterwards hauling the ash-pan. Note the average thickness of the firebed, quickly observe the steam pressure, water level, and time, and record the latter as the starting time. Immediately fire with weighed coal and increase the load as fast as may be to the working point, meanwhile establishing the furnace conditions determined on. Then proceed with the regular work of the test.

When the end of the test approaches, again burn down and clean the fire, meanwhile reducing the load as may be needed to prevent undue force of draft. As soon as the firebed reaches the condition observed at the start, immediately take the steam pressure, water level, and time, and record the latter as the stopping time. Finally haul the ashes and refuse from the ash-pan and remove the cinders from the smoke-box. If the firebed or water level at the end differs from that observed at the beginning, a correction is applied accordingly.

8 RECORDS

The general data should be recorded as pointed out in Part I, page 1703. Readings of the instruments concerned and a set of indicator diagrams should be obtained every ten minutes. A suitable number of additional diagrams should be taken during the operations of starting and stopping to obtain the average load for these short periods. Directions on the subject of indicating may be found on pages 1699 and 1736 and in Appendix No. 18.

9 SAMPLING AND DRYING COAL

During the progress of the trial the coal should be regularly sampled and its moisture determined in the manner pointed out on page 1712.

10 ASHES AND REFUSE

The ashes and refuse withdrawn from the ash-pan and smoke-box should be weighed in a dry state, and samples should be taken and analyzed if desired for unburned carbon.

11 CALORIFIC TESTS OF COAL

The quality of the coal should be determined by a calorimeter

test of the sample above referred to. Methods of making this test are described on page 1698 and in Appendix No. 12.

12 CALCULATION OF RESULTS

For methods of calculating the results, reference may be made to page 1722 of the Boiler Code and page 1735 of the Engine Code.

13 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object.

14 TESTS WITH OIL FUEL

When oil fuel is used the rules governing the tests may be modified to conform to the characteristics of liquid fuel. The "flying" method of starting and stopping may be employed, and the duration reduced.

TABLE 1 DATA AND RESULTS OF SHOP-TEST

CODE OF 1912

(1) Test of.....locomotive at.....
to determine.....conducted by.....
(2) Type of locomotive.....
(3) Type of auxiliaries.....
(4) Dimensions of cylinder.....
(5) Dimensions of boiler:
(*a*) Grate surface.....
(*b*) Water-heating surface.....
(*c*) Superheating surface.....
(6) Dimensions of auxiliaries.....
(7) Diameter of exhaust nozzles.....
(8) Date .....
(9) Duration .....hr.

AVERAGE PRESSURES AND TEMPERATURES

(10) Steam pressure in boiler by gage.....lb.
(11) Steam chest pressure.....lb.
(12) Temperature of feedwater.....deg.
(13) Temperature of smoke-box.....deg.
(14) Force of draft in smoke-box.....in.
(15) Force of draft in furnace.....in.

TOTAL QUANTITIES

(16) Total coal as fired.....lb.
(17) Moisture in coal.....per cent
(18) Total dry coal.....lb.
(19) Ash, cinders, and refuse:
  (*a*) From ash-pan.....lb.
  (*b*) From smoke-box.....lb.
  (*c*) Total.....lb.
(20) Percentage of ash, cinders, and refuse to dry coal.....per cent
(21) Percentage of ash by analysis.....per cent
(22) Total combustible consumed (Line 18—Line 19).....lb.
(23) Calorific value per lb. of dry coal by calorimeter test.....B.t.u.
(24) Total weight of water fed.....lb.
(25) Percentage of moisture in steam or number of degrees of superheating.....per cent or deg.
(26) Total water evaporated corrected for moisture or superheat in steam.....lb.
(27) Total equivalent evaporation from and at 212 deg.....lb.

HOURLY QUANTITIES

(28) Dry coal per hour.....lb.
(29) Dry coal per hour per sq. ft. of grate.....lb.
(30) Water evaporated per hour.....lb.
(31) Equivalent evaporation per hour from and at 212 deg.....lb.
(32) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water heating surface.....lb.

INDICATOR DIAGRAMS

(33) Mean effective pressure in lb. per sq. in.:
  (*a*) First cylinder.....
  (*b*) Second cylinder.....
(34) Back pressure at mid-stroke:
  (*a*) First cylinder.....
  (*b*) Second cylinder.....

SPEED

(35) Revolutions per minute.....rev.
(36) Corresponding miles per hour.....miles

POWER

(37) Indicated horsepower:
  (*a*) First cylinder.....i.h.p.
  (*b*) Second cylinder.....i.h.p.
  (*c*) Whole engine.....i.h.p.
(38) Pull on drawbar.....lb
(39) Dynamometer horsepower.....dyn.h.p.
(40) Brake horsepower.....br.h.p.

ECONOMY RESULTS

(41) Dry coal per i.h.p. per hour.....lb.
(42) Dry coal per dynamometer h.p.....lb.

(43) Dry steam per i.h.p. per hour.....lb.
(44) Dry steam per dynamometer h.p. hour.....lb.

BOILER PERFORMANCE

(45) Water evaporated per lb. of dry coal.....lb.
(46) Equivalent evaporation from and at 212 deg. per lb. of dry coal....lb.
(47) Equivalent evaporation from and at 212 deg. per lb. of combustible .....lb.
(48) Efficiency of boiler, furnace and grate.....per cent

CODE FOR ROAD TESTS

1 OBJECT AND PREPARATIONS

The leading object of a road test as covered by this code is the determination of the coal and steam consumption of a locomotive per unit of power under the practical conditions of railroad service. The preparations for a road test are the same as those noted for the shop test, except for restrictions imposed by the regular operation of the locomotive, and in addition those required for data to be obtained on the road. See Appendix No. 34.

It is assumed that a dynamometer car is available for registering the amount of pull on the drawbar, in the manner referred to in Appendix No. 20.

To facilitate the work of the men who operate the indicators and read the instruments at the front end of the locomotive, as also to protect them from wind and rain and jolting of the locomotive, the smoke-box should be surrounded with a suitable housing or pilot-box extending back to the cylinders.

2 FUEL[1]

Select the fuel required in accordance with the special object in view. If maximum efficiency or capacity is desired, the fuel should be some kind of coal that is regarded as a standard for the locality where the test is made, as noted in the Boiler Code, page 1715

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a road test of a locomotive are:

[1] See Section 14, Page 1781, in Code for Shop Tests for memoranda relating to oil fuel.

(*a*) Platform scales for weighing coal and ashes located at the terminals, the coal being weighed preferably into sacks duly marked
(*b*) A coal calorimeter
(*c*) A water meter suitably calibrated for measuring the feedwater consumption
(*d*) A graduated scale attached to the water glass, and one or more water glasses with suitable scales attached to the water tank of the tender
(*e*) Suitable levels or plumb-lines to show the inclination of the boiler and tender
(*f*) Pressure gages, draft gages, and thermometers
(*g*) A steam calorimeter
(*h*) Steam engine indicators
(*i*) A planimeter
(*j*) A tachometer or other speed-measuring apparatus
(*k*) A dynamometer car for determining the pull on the drawbar

Directions for their use and calibration are given on pages 1691 to 1702, pages 1713 and 1714, and in Appendix No. 34.

The steam used for auxiliary purposes other than the main engine, embracing air pump, blower, calorimeter, whistle, that escaping at the safety valve, etc., may be estimated from data obtained by testing them one by one either before or after the main trial, using the water-glass method described in Appendix No. 3.

4 OPERATING CONDITIONS

In a road test the operating conditions are those pertaining to the regular service on the railroad, embracing principally the usual stops at stations, the usual speeds, and other conditions required for making schedule time, including those pertaining to the methods of firing, manipulation of the throttle valve, reverse lever, etc. If it is desired to make the test under other conditions, these should all be determined and maintained accordingly.

5 DURATION

The duration of a road test depends upon the length of the route between terminals. In fast passenger service the route should, if practicable, be at least 100 miles long. In service requiring frequent stops and in freight service the distance may be much shorter.

The length of time upon which the hourly rates of combustion and evaporation are based is the total time that the throttle valve is open, and not the elapsed time between the so-called starting and stopping times.

6 STARTING AND STOPPING

The fire having been thoroughly cleaned and the ashes and refuse removed from the ash-pan and smoke-box, burn down the fire as low as practicable just before the locomotive leaves the round house, note its average thickness, the steam pressure, the water levels, the reading of the meter, and the time, and consider the latter the starting time. Thereafter cover the fire with weighed coal and proceed with the regular work of the test. The water data should be taken just before the train starts.

As the end of the route is approached, let the fire burn down to such an extent as to leave, as nearly as practicable, after cleaning, the same amount of live coal on the grate as that observed at the start. On reaching the terminal take the pressure, the water levels, the meter reading, and the time, and consider the latter the stopping time. Then clean the fire, and if its thickness differs from that observed at the start, make a correction in the weight of coal fired accordingly. Likewise, correct for any difference in the water levels. Finally haul the ashes and refuse from the ash-pan and remove the cinders and ash from the smoke-box.

7 RECORDS

The data should be recorded in the general manner pointed out on page 1703, bearing in mind the extremely fluctuating character of the load which often obtains, and the great variations in the class of service, whether passenger or freight. Readings of the instruments concerned, so far as they can be observed when the locomotive and train are in motion, should be taken every five minutes. Indicator diagrams should be obtained one after another as fast as possible, say a set every two minutes; and at the same time the corresponding pressure, position of throttle valve and reverse lever, speed, and force of draft should be observed. The intervals of time here given may be lengthened when the service is such that the conditions are substantially uniform. Special readings of the meter, water levels and total number of sacks of coal fired should be taken at each stopping place. Careful observations should be made throughout the trial of the time of passing each mile-post, the time that the throttle valve is opened and closed, not only at each stop but also when it is necessary to shut off steam on down grades, and the time of arriving and leaving each station; also the length of time the safety valve, whistle, blower, train heating system, lighting system

and other steam using apparatus are in operation. A record should be taken of the water levels before and after scooping water from track tanks the injector being stopped, and of the number of injector applications and water wasted.

8 SAMPLING AND DRYING COAL

While the coal is being weighed into the sacks, or otherwise weighed, it should be systematically sampled and its moisture determined in the manner pointed out on page 1712.

9 ASHES AND REFUSE

The ashes and refuse taken from the ash-pan and smoke-box at the close of the test should be weighed in a dry state, and a sample analyzed if desired for unburned carbon.

10 CALORIFIC TESTS OF COAL

The quality of the coal should be determined by a calorimeter test of the sample above referred to. Methods of making this test are described on page 1698 and in Appendix No. 12.

11 CALCULATION OF RESULTS

For methods of calculating the principal results, reference may be made to page 1722 of the Boiler Code and page 1735 of the Engine Code.

To determine the average drawbar pull, the dynamometer record should be averaged either by the use of a planimeter or by direct measurement at a sufficient number of points to give an equivalent result.

The maximum pull should be measured at the point of the record which shows the highest sustained pull, and not at the highest point which happens to be reached by the momentary fling of the marking pen.

12 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object:

TABLE 2 DATA AND RESULTS OF ROAD-TEST

CODE OF 1912

(1) Test of.....locomotive on.....
to determine.....conducted by.....
(2) Type of locomotive.....
(3) Type of auxiliaries.....
(4) Dimensions of cylinders.....
(5) Dimensions of boiler:
(*a*) Grate surface.....
(*b*) Water heating surface.....
(*c*) Superheating surface.....
(6) Dimensions of auxiliaries.....
(7) Diameter of exhaust nozzles.....
(8) Weight of locomotive and tender, including water.....tons
(9) Number of cars, including dynamometer car.....
(10) Weight of cars, including dynamometer car.....tons
(11) Length of route.....miles
(12) Train load expressed in ton-miles.....ton-miles
(13) Total load expressed in ton-miles.....ton-miles
(14) Schedule time between terminals.....hr.
(15) Name of engineman.....
(16) Name of fireman.....
(17) Date.....
(18) Duration (elapsed time between starting and stopping times....hr.
(19) Actual elapsed time between terminals.....hr.
(20) Length of time throttle valve is open.....hr.
(21) Number of stops.....
(22) Time consumed in stops.....hr.
(23) Running time (Line 19 — Line 22).....hr.

AVERAGE PRESSURES, TEMPERATURES, ETC.

(24) Steam pressure in boiler by gage.....lb.
(25) Steam chest pressure.....lb.
(26) Temperature of feedwater.....deg.
(27) Temperature of smoke-box.....deg.
(28) Force of draft in smoke-box.....in.
(29) Force of draft in furnace.....in.
(30) Weather, wind, etc.....
(31) Average position of throttle.....
(32) Average position of reversing lever.....
(33) Average number of strokes of air pump per minute.....

TOTAL QUANTITIES

(34) Total coal as fired.....lb.
(35) Moisture in coal.....per cent
(36) Total dry coal.....lb.
(37) Ash, cinders, and refuse:

(*a*) From ash-pan ..... lb.
(*b*) From smoke-box ..... lb.
(*c*) Total ..... lb.

(38) Percentage of ash, cinders, and refuse to dry coal ..... per cent
(39) Percentage of ash, by analysis ..... per cent
(40) Total combustible consumed (Line 36—Line 37) .....
(41) Calorific value per lb. of dry coal by calorimeter test ..... B.t.u.
(42) Total weight of water fed ..... lb.
(43) Percentage of moisture in steam or number of degrees of superheating ..... per cent. or deg.
(44) Total water evaporated corrected for moisture or superheat in steam ..... lb.
(45) Estimated weight of steam used by air pump, blower, calorimeter, whistle, that escaping at the safety valve, etc ..... lb.
(46) Percentage borne by steam used for auxiliary purposes named in Line 45 to total output ..... per cent

HOURLY QUANTITIES

(47) Dry coal per hour (Line 36 ÷ Line 20) ..... lb.
(48) Dry coal per hour per sq. ft. of grate ..... lb.
(49) Water evaporated per hour (Line 44 ÷ Line 20) ..... lb.
(50) Equivalent evaporation per hour from and at 212 deg ..... lb.
(51) Equivalent evaporation per hour from and at 212 deg. per sq. ft. of water heating surface ..... lb.
(52) Dry steam consumed by engine alone per hour (Line 44 — Line 45) ..... lb.

INDICATOR DIAGRAMS

(53) Mean effective pressure in lb. per sq. in.:
(*a*) First cylinder .....
(*b*) Second cylinder .....
(54) Back pressure at mid-stroke:
(*a*) First cylinder .....
(*b*) Second cylinder .....

SPEED

(55) Revolutions per minute ..... rev.
(56) Average speed in miles per hour ..... miles
(57) Maximum speed in miles per hour ..... miles

POWER

(58) Indicated horsepower:
(*a*) First cylinder ..... i.h.p.
(*b*) Second cylinder ..... i.h.p.
(*c*) Whole engine ..... i.h.p.
(59) Average drawbar pull ..... lb.
(60) Maximum drawbar pull ..... lb.
(61) Dynamometer horsepower ..... dyn.h.p.

ECONOMY RESULTS

(62) Dry coal consumed per i.h.p. per hour ..... lb.
(63) Dry coal consumed per dynamometer h.p. per hour ..... lb.

(64) Dry steam per i.h.p. per hour.....lb.
(65) Dry steam per dynamometer per h.p.-hr.....lb.
(66) Dry coal consumed per ton mile, train load and total load.....lb.

BOILER PERFORMANCE .

(67) Water evaporated per lb. of dry coal.....lb.
(68) Equivalent evaporation from and at 212 deg. per lb. of dry coal...lb.
(69) Equivalent evaporation from and at 212 deg. per lb. of combustible .....lb.
(70) Efficiency, based on dry coal.....per cent

PART XII

GAS PRODUCER CODE

RULES FOR CONDUCTING TESTS OF GAS PRODUCERS

1 OBJECT AND PREPARATIONS

Determine the object, take the dimensions, note the physical condition of the producer and its appurtenances, install the testing appliances, etc., as pointed out in the general instructions given on pages 1689, 1690, 1691, 1702 and 1703, and make preparations for the test accordingly.

2 FUEL[1]

Determine the character of the fuel to be used. If an untried fuel is selected and a test-producer is available, make a preliminary trial of the fuel in this apparatus and ascertain its working characteristics and the proper methods of handling it.

In tests of maximum efficiency and capacity of a producer for comparison with other producers, the fuel should be some kind of coal which is commercially regarded as a standard for such use in the locality where the test is made. The coal selected for such tests should be the best of its class and free from unusual slag-forming impurities.

The size of the coal should be determined by screening a sample, using the screens referred to on page 1692, or in Appendix No. 7.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for producer tests are

(*a*) Platform scales for weighing coal and ashes
(*b*) A coal calorimeter
(*c*) A gas calorimeter
(*d*) Gas analyzing apparatus and appliances for determining tar and soot

[1] This code is primarily intended for producers using coal. If other fuel, such as wood or oil is burned, the rules may be modified accordingly.

(*e*) A gas meter, pitot tube, or other suitable apparatus for measuring the gas output

(*f*) A manometer or pressure gage

(*g*) Water meters for measuring feed and scrubber water, and steam meters for measuring steam used by the apparatus

(*h*) Thermometers

Full directions regarding the use and calibration of the above noted appliances are given on pages 1691 to 1702, and in various appendices there referred to.

The location of the pitot tube, if used, should be in the delivery pipe at a point near the producer or just beyond the scrubber, or at both points, according to the use made of the gas, either for fuel or power, and other requirements.

4 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view, as pointed out on page 1703, and see that they prevail throughout the trial.

5 DURATION

The duration of both efficiency and capacity tests of a producer, with the exceptions noted below, should be such that the total consumption of fuel is at least ten times the weight of the fuel contained in the producer when in normal operation, estimating this weight in the case of coal at 45 lb. per cu. ft.

In cases like down-draft producers which require the fuel bed to be entirely removed and rebuilt at regular intervals, and in producers where a complete cleaning and renewal occurs before the total consumption above stipulated has been reached, the duration should be that of the regular commercial operating cycle, or the time elapsing between two successive renewals of the fuel bed.

6 STARTING AND STOPPING

The conditions regarding the temperature of the producer and its contents, and the quantity and quality of the latter, should be as nearly as possible the same at the end as at the beginning of the trial. To secure the desired equality of conditions, the starting and stopping should occur at times of regular cleanings, and they should be preceded for a period of not less than 10 hours by the same regular

working conditions as those characterizing the test as a whole. The operations of starting and stopping should then be carried on as follows:

(*a*) *Up-Draft Suction-Producers.* Remove the ash and clinkers from the grate and the lower part of the furnace space, taking care that the crust or closely-united layer which supports the coal above is not unduly disturbed. Then break open the crust and allow the mass to drop into the space left vacant. Introduce a poker rod through the poke holes in the upper head and stir up the coal within, thereby causing it to settle and fill the remaining spaces. As a final step, quickly replenish the producer with coal, leaving the hopper level-full, take the time, and consider this the starting time. Then clean the ash-pit, and thereafter proceed with the regular work of the test, using weighed coal.

When the time arrives for bringing the trial to a close, the cleaning operations described above are repeated, ending with filling the hopper, taking the time, and considering this the stopping time; finally hauling the ashes and refuse from the ashpit.

(*b*) *Up-Draft Pressure Producers.* Remove the ashes until the top of the ashbed is lowered to the normal working point, say six inches above the blast-hood. Introduce the poker-rod and break down any bridge or crust that may have formed, at the same time closing up the channels that run through the fuel bed, thereby making the bed homogeneous. Then replenish the producer with coal, fill the hopper level-full, take the time, and consider this the starting time. Thereafter proceed with the regular work of the test, using weighed coal.

When the time approaches for closing the test, the operations above described are repeated, ending with replenishing the producer and filling the hopper with weighed coal, taking the time, and considering this the stopping time. The ashes and refuse finally removed are to be dried before weighing, or a sample should be taken and the moisture, as determined therefrom, allowed for.

(*c*) *Down-Draft Pressure Producer.* Thoroughly clean the producer of its entire contents. Introduce a weighed supply of coke or coal, start the fire, and build up the fuel bed to its working condition, using weighed coal. When this point is reached, take the time, and consider this the starting time. Thereafter proceed with the regular work of the test.

When the time approaches for closing the test, burn the fuel bed as low as practicable to prepare for cleaning, stop the exhauster, note the time, and consider this the stopping time. Then completely empty the producer, quench the fire remaining in the live coals, separate and weigh the coke and ash, and deduct the weight of the former from that of the coke as charged. Finally dry the ash and refuse, or take a sample and allow for the moisture determined therefrom.

7 RECORDS

8 SAMPLING AND DRYING COAL

9 ASHES AND REFUSE

10 CALORIFIC TESTS AND ANALYSES OF COAL

The directions pertaining to the above divisions of the subject are practically the same as those given under the corresponding headings in the Boiler Code, and reference may be made to pages 1720 and 1721 of that code for these directions.

11 CALORIFIC TESTS AND ANALYSES OF GAS OUTPUT

The quality of the gas should be determined by calorific tests and analyses, continuous samples for this purpose being taken from the delivery pipe at a point near the producer and at other points as may be needed.

The calorific test should be made with the Junker calorimeter, described in Appendix No. 13, or its equivalent. Unless otherwise required the "higher value" should be employed in calculating the results of the test. For an approximate determination of the composition of the gas, the Orsat apparatus may be used, and for complete determinations, the Hempel apparatus or its equivalent. Both of the two named are described in Appendix No. 15. The frequency with which these determinations should be made depends on the uniformity of the output, but the intervals, where practicable, should not be more than one-half hour, the time taken for collecting each sample being not less than one-half hour.

12 CALCULATION OF RESULTS

(*a*) *Total Volume of Gas Delivered.* The volume of gas found by pitot tube measurement is determined by multiplying the area of the delivery pipe in sq. ft. at the tube by the velocity of the gas in ft. per minute, and the product by the duration of the trial in minutes. The equivalent volume at atmospheric pressure (30 in. barometer) and temperature of 62 deg. fahr., is obtained by multiplying the absolute pressure of the gas in lb. per sq. in. (gage pressure plus 14.7) by the constant 35.3, and dividing the product by the absolute temperature of the gas (temperature by thermometer plus 460 deg.).

(*b*) *Net Volume of Gas Delivered.* The net volume of gas delivered is found by subtracting from the total volume the equivalent volume of gas required for furnishing steam drawn from an outside source, if any, or for furnishing power used for any purpose concerned in the operation of the producer and its auxiliaries.

(*c*) *Weight of Gas.* The weight of dry gas delivered is found by multiplying the volume in cu. ft., reduced to 62 deg. and 30 in., given in Table 2, Line 21, by the weight per cu. ft. of gas given in Table 2, Line 77.

The weight of the gas per cu. ft. is determined by multiplying the percentage of each component gas as found by analysis (see Lines 55 to 63, Table 2) by its weight in lb. per cu. ft., at 62 deg. and 30 in., as given in the following table, and dividing the sum of the products by 100.

CO_2	0.1116	CH_4	0.0428
CO	0.0736	C_2H_4	0.0737
O	0.0842	SO_2	0.1638
H	0.0053	H_2S	0.0868
N	0.0740		

(*d*) *Moisture in Gas Leaving Producer.* The percentage of moisture in the gas is found by passing a measured sample of the gas through a chloride of calcium tube and weighing the amount of moisture absorbed.

(*e*) *Percentage of Tar and Soot in Gas.* The percentage of tar and soot is found by comparing the total weight determined, including that collected from the various tar drips with the total weight of dry fuel used.

(*f*) *Efficiency.* The efficiency is the relation between the calorific value of the gas per lb. of fuel charged, or combustible burned, and the calorific value of 1 lb. of fuel or combustible. The former is ascertained by multiplying the B.t.u. per cu. ft. of gas as determined by the calorimeter test (higher value) by the cu. ft. of gas delivered, and dividing the product by the total weight of fuel charged or combustible burned.

The "combustible burned" is determined by subtracting from the weight of coal charged the moisture in the coal and the ash and refuse, including unburned coal which is withdrawn from the producer or ash-pit during the progress of the trial. The "combustible" used for determining the calorific value is the weight of the coal less the moisture and ash found by analysis.

The efficiency of "conversion and cleaning" in the above calculation is found by using the total volume of gas delivered. The "efficiency of the plant" is found by using the net volume of gas delivered.

(*g*) *Heat Balance.* The various quantities showing the distribution of heat in the heat balance given in Table 2, are computed in the following manner:

The heat contained in the dry gas is found by multiplying the cubic feet of gas at 62 deg. and 30 in. per lb. of dry coal by the calorific value of 1 cu. ft. of gas at 62 deg. and 30 in. (higher value).

The heat carried away by the scrubber is obtained by multiplying the weight of water fed to the scrubber by the number of degrees rise

of temperature, and dividing the product by the total weight of dry coal consumed.

The heat contained in the moisture leaving the producer is found by multiplying the total weight of dry gas by the proportion of moisture in the gas leaving the producer and by the total heat of 1 lb. of superheated steam at the temperature of the gas leaving the producer reckoned from the temperature of the air in the room, and dividing the product by the weight of dry coal consumed.

13 DATA AND RESULTS

The data and results should be reported in accordance with either the Short Form or Complete Form given herewith, adding lines for data not provided for or omitting those not required as may conform to the object in view.

If a preliminary trial of the fuel is made in a test-producer, add to the table the general results obtained.

14 CHART

In trials having for an object the determination and exposition of the complete performance from beginning to end, the entire log of readings and data should be plotted on a chart and represented graphically.

TABLE 1 DATA AND RESULTS OF GAS PRODUCER TEST

SHORT FORM, CODE OF 1912

(1) Test of.....producer located at.....
to determine.....conducted by.....
(2) Type of producer.....
(3) Rated capacity of producer.....
(4) Date .....
(5) Duration .....hr.
(6) Kind of coal[1] and where mined.....
(7) Size of coal.....

AVERAGE PRESSURES, TEMPERATURES, ETC.

(8) Steam pressure in vaporizer by gage.....lb.
(9) Gas pressure in delivery main at point where gas is measured.....in. water
(10) Temperature of feedwater.....deg.

[1] If other fuel than coal is used, the lines may be changed to read accordingly.

(11) Temperature of gas in delivery main near producer.....deg.
(12) Temperature of gas in delivery main at point where gas is measured .....deg.
(13) Force of blast or draft in ashpit.....in.

TOTAL QUANTITIES.

(14) Weight of coal as charged.....lb.
(15) Percentage of moisture in coal.....per cent
(16) Total weight of dry coal consumed.....lb.
(17) Total ash and refuse.....lb.
(18) Percentage of ash and refuse in dry coal.....per cent
(19) Total cu. ft. of gas as measured.....cu. ft.
(20) Equivalent cu. ft. of gas at 62 deg. and 30 in.....cu. ft.
(21) Net cu. ft. of gas at 62 deg. and 30 in.[1].....cu. ft.
(22) Total water fed to vaporizer.....lb.
(23) Total water supplied to scrubber.....lb.

HOURLY QUANTITIES

(24) Dry coal consumed per hour.....lb.
(25) Dry coal per sq. ft. of main fuel bed per hour.....lb.
(26) Total cu. ft. of gas delivered per hour.....cu. ft.
(27) Total cu. ft. of gas per hour at 62 deg. and 30 in.....cu. ft.
(28) Net cu. ft. of gas per hour at 62 deg. and 30 in.....cu. ft.

ECONOMY RESULTS

(29) Total cu. ft. of gas delivered per lb. of dry coal (Line 13 ÷ Line 10).....cu. ft.
(30) Equivalent total gas at 62 deg. and 30 in. per lb. of dry coal...cu. ft.
(31) Net cu. ft. of gas at 62 deg. and 30 in. per lb. of dry coal[1].....cu. ft.
(32) Net cu. ft. of gas at 62 deg. and 30 in. per lb. of combustible...cu. ft.

EFFICIENCY[2]

(33) Calorific value of dry coal per lb.....B.t.u.
(34) Calorific value of combustible per lb.....B.t.u.
(35) Calorific value of gas per cu. ft. (higher value).....B.t.u.
(36) Efficiency of producer based on coal.....per cent
(37) Efficiency of producer based on combustible.....per cent

COST OF PRODUCTION

(38) Cost of coal per ton of .... lb. delivered.....dollars
(39) Cost of coal required for producing 10,000 net cu. ft. of gas at 62 deg. and 30 in.....dollars
(40) Cost of coal required for producing one million B.t.u. in the gas .....dollars

[1] After deducting equivalent gas required for auxiliaries.

[2] If the efficiency is based on the "low value" of the heat units in the gas the fact should be so stated.

TABLE 2 DATA AND RESULTS OF GAS PRODUCER TEST

COMPLETE FORM, CODE OF 1912

(1) Test of.....producer located at.....
to determine .....conducted by.....

DIMENSIONS

(2) Outside diameter of producer.....ft.
(3) Height of producer.....ft.
(4) Inside diameter of producer.....ft.
(5) Diameter of grate.....ft.
(6) Area of grate.....sq. ft.
(7) Percentage of air space in grate.....per cent
(8) Area of fuel bed (at maximum diameter).....sq. ft.
(9) Area of water heating surface in vaporizer.....sq. ft.
(10) Rated capacity of producer in lb. of coal per hour.....lb.
(11) Date .....
(12) Duration .....hr.
(13) Kind of coal[1] and where mined.....
(14) Size of coal.....

AVERAGE PRESSURES, TEMPERATURES, ETC.

(15) Steam pressure in vaporizer by gage.....lb.
(16) Gas pressure in main at point where gas is measured.....in. water
(17) Force of blast or draft in ashpit.....in. water
(18) Barometric pressure.....in. merc.
(19) Temperature of feedwater entering vaporizer.....deg.
(20) Temperature of gas in main near producer.....deg.
(21) Temperature of gas in main at point where gas is measured.....deg.
(22) Temperature of air in room.....deg.
(23) Temperature of water entering scrubber.....deg.
(24) Temperature of water leaving scrubber.....deg.
(25) Weight of dry gas per cu. ft. reduced to 62 deg. and 30 in.....lb.

TOTAL QUANTITIES

(26) Weight of coal as fired.....lb.
(27) Percentage of moisture in coal.....per cent
(28) Total weight of dry coal consumed.....lb.
(29) Total ash and refuse.....lb.
(30) Percentage of ash and refuse in dry coal.....per cent
(31) Total number of cu. ft. of gas as measured.....cu. ft.
(32) Equivalent cu. ft. of gas at temperature of 62 deg. and pressure of atmosphere of 30 in.....cu. ft.
(33) Net cu. ft. of gas at 62 deg. and 30 in.[2].....lb.
(34) Total weight of dry gas.....lb.
(35) Percentage of moisture in gas leaving producer.....per cent

[1] If other fuel than coal is used the lines may be changed to read accordingly.
[2] After deducting equivalent gas required for auxiliaries.

(36) Percentage of tar and soot in gas referred to total fuel....per cent
(37) Total water fed to vaporizer.....lb.
(38) Total water evaporated in vaporizer.....lb.
(39) Total weight of steam supplied to producer.....lb.
(40) Total weight of water fed to scrubber.....

HOURLY QUANTITIES

(41) Dry coal consumed per hour.....lb.
(42) Dry coal consumed per hour per sq. ft. of grate.....lb.
(43) Dry coal consumed per hour per sq. ft. of main fuel bed.....lb.
(44) Total cu. ft. of gas delivered per hour (Line 20 ÷ Line 17)...cu. ft.
(45) Total cu. ft. of gas per hour at 62 deg. and 30 in.....cu. ft.
(46) Net cu. ft. of gas delivered per hour at 62 deg. and 30 in.....cu. ft.
(47) Weight of dry gas per hour.....lb.
(48) Water fed per hour to vaporizer.....lb.
(49) Water evaporated per hour in vaporizer.....lb.
(50) Steam supplied to producer per hour.....lb.
(51) Water fed to scrubber per hour.....lb.

PROXIMATE ANALYSIS OF COAL

(52) Fixed carbon.....per cent
(53) Volatile matter.....per cent
(54) Moisture .....per cent
(55) Ash .....per cent

100 per cent

(56) Sulphur, separately determined.....per cent

ULTIMATE ANALYSIS OF DRY COAL

(57) Carbon (C).....per cent
(58) Hydrogen (H).....per cent
(59) Oxygen (O).....per cent
(60) Nitrogen (N).....per cent
(61) Sulphur (S).....per cent
(62) Ash .....per cent

100 per cent

(63) Moisture in sample of coal as received.....per cent

ANALYSIS OF ASH AND REFUSE

(64) Carbon .....per cent
(65) Earthy matter.....per cent

ANALYSIS OF GAS BY VOLUME

(66) Carbon dioxide (CO_2).....per cent
(67) Carbon monoxide (CO).....per cent
(68) Oxygen (O).....per cent
(69) Hydrogen (H).....per cent
(70) Marsh gas (CH_4).....per cent
(71) Olefiant gas (C_2H_4).....per cent
(72) Sulphur dioxide (SO_2).....per cent
(73) Hydrogen sulphide (H_2S).....per cent
(74) Nitrogen (N by difference).....per cent

100 per cent

Calorific Values by Calorimeter

(75) Calorific value of dry coal per lb.....B.t.u.
(76) Calorific value of combustible per lb.....B.t.u.
(77) Calorific value of gas per cu. ft. at 62 deg. and 30 in. (higher value).....B.t.u.

Economy Results

(78) Total cu. ft. of gas as measured, per pound of dry coal consumed.....cu. ft.
(79) Equivalent cu. ft. of gas at 62 deg. and 30 in. per lb. of dry coal.cu. ft.
(80) Equivalent cu. ft. of gas at 62 deg. and 30 in. per lb. of combustible.....cu. ft.
(81) Net cu. ft. of gas at 62 deg. and 30 in. per lb. of dry coal.....cu. ft.
(82) Net cu. ft. of gas at 62 deg. and 30 in. per lb. of combustible..cu. ft.

Efficiency[1]

(83) Efficiency of producer based on coal:
  (*a*) Conversion and cleaning.....per cent
  (*b*) Plant.....per cent
(84) Efficiency of producer based on combustible:
  (*a*) Conversion and cleaning.....per cent
  (*b*) Plant.....per cent

Cost of Production

(85) Cost of coal per ton of .... lb., delivered.....dollars
(86) Cost of coal required for producing 10,000 net cu. ft. of gas at 62 deg. and 30 in.....dollars
(87) Cost of coal for producing 1,000,000 B.t.u.....dollars

Heat Balance Based on 1 Lb. of Dry Coal

		B.t.u.	Per cent
(88)	Heat contained in dry gas	.....	.....
(89)	Heat carried away by scrubber	.....	.....
(90)	Heat contained in moisture leaving producer	.....	.....
(91)	Heat unaccounted for, including radiation,—difference between the sum of Lines 88, 89, and 90 and Line 92	.....	.....
(92)	Total calorific value of 1 lb. of dry coal, same as Line 75	.....	.....

[1] If the efficiency is based on the "low value" of the heat units in the gas, the fact should be so stated.

PART XIII

GAS AND OIL ENGINE CODE

RULES FOR CONDUCTING TESTS OF GAS AND OIL ENGINES

1 OBJECT AND PREPARATIONS

Determine the object, take the dimensions, note the physical condition of the engine and its appurtenances, install the testing appliances, etc., as explained in the general instructions given on pages 1689, 1690, 1691, 1702 and 1703, and make preparations for the test accordingly.

2 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for simple performance tests of gas and oil engines are:

(*a*) Tanks and platform scales for weighing oil
(*b*) A calorimeter for determining the heat of combustion of oil
(*c*) A gas meter or other apparatus for measuring gas
(*d*) A gas calorimeter
(*e*) Pressure gages and thermometers
(*f*) Gas engine indicators
(*g*) A planimeter
(*h*) A tachometer or other speed-measuring apparatus

and in addition for the determination of the complete performance

(*i*) Gas analyzing apparatus
(*j*) A water meter for measuring jacket water
(*k*) A friction brake or dynamometer

Full directions regarding the use and calibration of these appliances are given on pages 1691 to 1702, and in various appendices there referred to.

3 OPERATING CONDITIONS

Determine what the operating conditions should be to conform to the object in view, and see that they prevail throughout the trial, as pointed out on page 1703, Part I.

4 DURATION

The test of a gas or oil engine with substantially constant load should be continued for such time as may be necessary to obtain a

number of successive records covering periods of half an hour or less during which the results are found to be uniform. In such cases a duration of three to five hours is sufficient for all practical purposes.

5 STARTING AND STOPPING

The engine having been set to work under the prescribed conditions and thoroughly heated (except in cases where the object is to obtain the performance under working conditions), the test is begun at a certain predetermined time by commencing to weigh the oil, or measure the gas, as the case may be, and take other data concerned; after which the regular measurements and observations are carried forward until the end. When the stopping time arrives the test is closed by simply taking the final readings.

6 RECORDS

The general data should be taken and recorded in the same manner as that described in the Steam Engine Code, on page 1735, to which reference may be made.

7 CALORIFIC TESTS AND ANALYSES

The quality of the oil or gas should be determined by calorific tests and analyses made on representative samples.

Directions for these tests or analyses are given on pages 1697 and 1698 under the headings Coal Calorimeters, Gas Calorimeters, Coal Analysis Apparatus, and Gas Analysis Apparatus, and in Appendices Nos. 12, 13, 14, and 15 there referred to.

8 CALCULATION OF RESULTS

(*a*) *Heat Consumption.* The number of heat units consumed by the engine is found by multiplying the heat units per lb. of oil or per cu. ft. of gas (higher value), as determined by calorimeter test, by the total weight of oil in lb. or volume of gas in cu. ft. consumed during the trial.

(*b*) *Horsepower and Efficiency.* The indicated horsepower, brake horsepower, and efficiency are computed by the same methods as those explained in the Steam Engine Code, on page 1736, to which reference may be made.

(*c*) *Heat Balance.* The various quantities showing the distribution of heat in the heat balance given in Table 2 are computed in the following manner:

The heat converted into work per i.h.p.-hr. (2545 B.t.u.) is found by dividing the work representing 1 h.p., or 1,980,000 ft-lb., per hour by the number of ft.-lb. representing 1 B.t.u., or 778.

The heat rejected in the cooling water is obtained by multiplying the weight of water supplied by the number of degrees rise of temperature, and dividing the product by the indicated horsepower.

The heat rejected in the dry exhaust gases per i.h.p.-hr. is found by multiplying the weight of these gases per i.h.p-hr. by the sensible heat of the gas reckoned from the temperature of the air in the room and by its specific heat. The weight of the dry gases per i.h.p.-hr. may be found by multiplying the weight of dry gas per lb. of carbon, using the formula:

$$\frac{11\,CO_2 + 8\,O + 7(CO + N)}{3(CO_2 + CO)}$$

in which CO_2, CO, O, and N are expressed in percentages by volume, by the proportion borne by the carbon in the total fuel, (either gas or oil), and by the weight of fuel per i.h.p-hr.

The heat lost in the moisture formed by the burning of hydrogen in the gas is found by multiplying the total heat of 1 lb. of superheated steam at the temperature of the gas, reckoning from the temperature of the air in the room, by the proportion of the hydrogen in the fuel as determined from the analysis, and multiplying the result by 9.

The heat lost through incomplete combustion is obtained by analyzing the exhaust gases and computing the heat of the unburned products which would have been produced by their combustion.

9 DATA AND RESULTS

The data and results should be reported in accordance with either the Short Form or Complete Form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view[1]:

TABLE 1 DATA AND RESULTS OF GAS OR OIL ENGINE TEST

SHORT FORM, CODE OF 1912

(1) Test of.....engine, located at.....
to determine..... conducted by.....
(2) Type and class of engine and number of cycles.....
(3) Dimensions:
(*a*) Single or double acting.....
(*b*) Diameter of cylinders.....in.
(*c*) Stroke of pistons.....ft.
(*d*) Diameter of piston rods.....in.
(*e*) Compression space or clearance.....per cent
(*f*) H.p. constant for 1 lb. m.e.p. and 1 r.p.m.....

[1] See note page 1744 for reference to engines driving electric generators and other machinery.

(4) Rated capacity .....
(5) Date .....
(6) Duration .....hr.
(7) Kind of gas.....
(8) Kind of oil.....
(9) Physical properties of oil (specific gravity, burning point, and flashing point).....

AVERAGE PRESSURES AND TEMPERATURES

(10) Pressure of gas near meter.....in.
(11) Barometric pressure.....lb.
(12) Temperature of cooling water:
(*a*) Inlet .....deg.
(*b*) Outlet .....deg.
(13) Temperature of gas near meter.....deg.
(14) Temperature of air.....deg.
(15) Temperature of exhaust gases.....deg.

TOTAL QUANTITIES

(16) Gas or oil consumed.....cu. ft.-lb.
(17) Cooling water supplied to jackets.....lb.
(18) Calorific value of gas per cu. ft., or of oil per lb., by calorimeter test (higher value).....b.t.u.

HOURLY QUANTITIES

(19) Gas or oil consumed per hour.....cu. ft.-lb.
(20) Cooling water supplied to jackets per hour.....lb.

INDICATOR DIAGRAMS

(21) Maximum pressure .....lb.
(22) Mean effective pressure.....lb.

SPEED AND EXPLOSIONS

(23) Revolutions per minute.....rev.
(24) Average number of explosions per minute.....

POWER

(25) Indicated horsepower.....i.h.p.
(26) Brake horsepower.....br. h.p.
(27) Friction horsepower by difference Line 25 — Line 26.....fr. h.p.
(28) Friction horsepower by friction diagrams.....fr. h.p.
(29) Percentage of indicated horsepower lost in friction (Line 27), per cent

ECONOMY RESULTS

(30) Heat units consumed by engine per hour[1]:
(*a*) Per indicated horsepower .....B.t.u.
(*b*) Per brake horsepower .....B.t.u.
(31) Pounds of oil or cubic feet of gas consumed per hour:
(*a*) Per indicated horsepower .....lb-cu. ft.
(*b*) Per brake horsepower .....lb-cu. ft.

SAMPLE DIAGRAMS

[1] If these results in the case of a gas engine are based on the "low value" of the heat units, that fact should be so stated.

NOTE: For an engine driving an electric generator the form may be enlarged to include electrical data in the manner given in the Steam Turbine Code, page 1749.

TABLE 2 DATA AND RESULTS OF GAS OR OIL ENGINE TEST

COMPLETE FORM, CODE OF 1912

(1) Test of.....engine, located at.....
to determine....., conducted by.....
(2) Type of engine, whether oil or gas.....
(3) Class of engine (mill, marine, motor for vehicle, pumping, or other).....
(4) Number of revolutions for one cycle, and class of cycle.....
(5) Method of ignition.....
(6) Name of builders.....
(7) Dimensions:
(*a*) Class of cylinder, whether working or compressing.....
(*b*) Vertical or horizontal.....
(*c*) Single or double acting.....
(*d*) Diameter of cylinders.....in.
(*e*) Stroke of pistons.....ft.
(*f*) Diameter of piston rods.....in.
(*g*) Compression space or clearance.....
(*h*) H.p. constant for 1 lb. m.e.p. and 1 r.p.m.....
(8) Rated capacity .....
(9) Date .....
(10) Duration .....hr.
(11) Kind of oil.....
(12) Physical properties of oil (specific gravity, burning point, flashing point).....
(13) Kind of gas.....

AVERAGE PRESSURES AND TEMPERATURES

(14) Pressure of gas near meter.....in.
(15) Barometric pressure.....in.
(16) Temperature of cooling water:
(*a*) Inlet .....deg.
(*b*) Outlet .....deg.
(17) Temperature of gas near meter.....deg.
(18) Temperature of air:
(*a*) By dry-bulb thermometer.....deg.
(*b*) By wet-bulb thermometer.....deg.
(19) Temperature of exhaust gases.....deg.

TOTAL QUANTITIES

(20) Gas or oil consumed.....cu. ft.-lb.
(21) Air supplied in cu. ft.....cu. ft.
(22) Cooling water supplied to jackets.....lb.
(23) Calorific value of oil per lb., or of gas per cu. ft., by calorimeter test (higher value).....B.t.u.

HOURLY QUANTITIES

(24) Gas or oil consumed per hour.....cu. ft-lb.
(25) Cooling water supplied per hour.....lb.

ANALYSIS OF OIL

(26) Carbon (C) ..... per cent
(27) Hydrogen (H) ..... per cent
(28) Oxygen (O) ..... per cent
(29) Sulphur (S) ..... per cent
(30) Moisture ..... per cent

100 per cent

ANALYSIS OF GAS BY VOLUME

(31) Carbon dioxide (CO_2) ..... per cent
(32) Carbon monoxide (CO) ..... per cent
(33) Oxygen (O) ..... per cent
(34) Hydrogen (H) ..... per cent
(35) Marsh gas (CH_4) ..... per cent
(36) Olefiant gas (C_2H_4) ..... per cent
(37) Sulphur dioxide (SO_2) ..... per cent
(38) Hydrogen sulphide (H_2S) ..... per cent
(39) Nitrogen (N by difference) ..... per cent

100 per cent

INDICATOR DIAGRAMS

(40) Pressure in lb. per sq. in. above atmosphere:
  (*a*) Maximum pressure ..... lb.
  (*b*) Pressure at beginning of stroke ..... lb.
  (*c*) Pressure at end of expansion ..... lb.
  (*d*) Exhaust pressure at lowest point ..... lb.
(41) Mean effective pressure in lb. per sq. in. ..... lb.

SPEED AND EXPLOSIONS

(42) Revolutions per minute ..... rev.
(43) Average number of explosions per minute .....
(44) Variation of speed between no load and full load ..... rev.
(45) Fluctuation of speed on suddenly changing from full load to no load measured by the increase in the revolutions due to the change .....

POWER

(46) Indicated horsepower ..... i.h.p.
(47) Brake horsepower ..... br.h.p.
(48) Friction horsepower by difference (Line 46 — Line 47) ..... fr. h.p.
(49) Friction horsepower by friction diagrams ..... fr. h.p.
(50) Percentage of indicated horsepower lost in friction (Line 48), per cent

Economy Results

(51) Heat units consumed by engine per hour[1]:
- (*a*) Per indicated horsepower.....B.t.u.
- (*b*) Per brake horsepower.....B.t.u.

(52) Pounds of oil or cubic feet of gas consumed per hour:
- (*a*) Per indicated horsepower.....lb-cu. ft.
- (*b*) Per brake horsepower.....lb-cu. ft.

Efficiency

(53) Thermal efficiency ratio:
- (*a*) Per indicated horsepower.....per cent
- (*b*) Per brake horsepower.....per cent

Work Done Per Heat Unit

(54) Ft-lb. of net work per b.t.u. consumed (1,980,000 ÷ Line 51*b*)...ft-lb.

Heat Balance Based on B.t.u. per i.h.p. per Hour

		B.t.u.	Per Cent
(55)	Heat converted into work.....	2545	.....
(56)	Heat rejected in cooling water.....	.....	.....
(57)	Heat rejected in the exhaust gases.....	.....	.....
(58)	Heat lost due to moisture formed by burning of hydrogen	.....	.....
(59)	Heat lost by incomplete combustion.....	.....	.....
(60)	Heat unaccounted for, including radiation.....	.....	.....
(61)	Total heat consumed per i.h.p. per hour, same as Line 51*a*.....	.....	.....

Sample Diagrams

[1] If these results, in the case of a gas engine, are based on the low value of the heat units, that fact should be so stated.

Note: For an engine driving an electric generator, the form may be enlarged to include electrical data in the manner given in the Steam Turbine Code, page 1749.

PART XIV

CODE FOR GAS POWER PLANTS

RULES FOR CONDUCTING TESTS OF COMPLETE GAS POWER PLANTS

1 OBJECT AND PREPARATIONS

The usual object of testing a complete gas power plant, embracing producer, engine, and appurtenances, is the determination of its commercial performance, i. e., the number of lb. of fuel consumed per unit of work done in a unit of time, and the rules given in this code apply to tests having that object. For directions pertaining to tests of the producer and engine individually, reference may be made to the Producer and Gas Engine Codes, pages 1791 and 1803.

Read the general instructions given on pages 1689, 1690, 1691, 1702 and 1703, take the dimensions, note the physical conditions of the plant, install the testing appliances, etc., as there explained, and prepare for the test accordingly.

2 FUEL

Determine the character of the fuel to be used as pointed out in the Gas Producer Code, page 1791.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a simple performance test of a gas power plant are:

(*a*) Platform scales for weighing coal and ashes
(*b*) A coal calorimeter
(*c*) Gas engine indicators
(*d*) A tachometer or other speed-measuring apparatus
(*e*) Wattmeters, in the case of an electric plant

Directions regarding the use and calibration of these appliances are given on pages 1691 to 1702.

If the test involves the determination of the individual performance of the producer and engine, additional instruments are required as pointed out in the Producer and Gas Engine Codes.

4 OPERATING CONDITIONS

Determine what the operating conditions should be as explained on page 1703 and see that they prevail throughout the trial.

5 DURATION

The duration of a gas producer plant test should conform to that of the producer alone, rules pertaining to which may be found in the gas producer code.

In cases where the engine is in operation only a part of the day, the hourly consumption of coal from which the economy results are computed should be the total coal burned in the producer divided by the number of hours that the engine is in operation at its working speed.

6 STARTING AND STOPPING

The rules for starting and stopping a complete plant test are governed by those required for starting and stopping the test of the producer, which are those given in the Producer Code, page 1791.

7 RECORDS

8 SAMPLING AND DRYING COAL

9 ASHES AND REFUSE

10 CALORIFIC TESTS AND ANALYSES OF COAL

11 CALCULATION OF RESULTS

The directions pertaining to the above five divisions of the subject are practically the same as those given under the corresponding headings in the Code for Complete Steam Power Plants, pages 1753 and 1754.

12 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for, or omitting those not required, as may conform to the object in view.

DATA AND RESULTS OF TEST OF COMPLETE GAS POWER PLANT

CODE OF 1912

(1) Test of.....gas power plant at.....
to determine....., conducted by.....
(2) Type and dimensions of producers.....
(3) Rated capacity of producers.....
(4) Total area of main fuel bed at maximum diameter.....sq. ft.
(5) Type and dimensions of engine.....
(6) Rated power of engine.....
(7) Date .....
(8) Duration .....hr.
(9) Kind of coal.....
(10) Size of coal.....

AVERAGE PRESSURES AND TEMPERATURES

(11) Pressure of gas near throttle valve.....in.
(12) Barometric pressure.....in.
(13) Temperature of cooling water leaving engine.....deg.
(14) Temperature of air in room.....deg.

TOTAL QUANTITIES

(15) Weight of coal as charged.....lb.
(16) Percentage of moisture in coal.....per cent
(17) Total weight of dry coal consumed.....lb.
(18) Total ash and refuse.....lb.
(19) Percentage of ash and refuse in dry coal.....per cent
(20) Calorific value of 1 lb. of dry coal by calorimeter test.....B.t.u.
(21) Cost of coal per ton of .... lb.....dollars

HOURLY QUANTITIES

(22) Dry coal consumed per hour.....lb.
(23) Dry coal per sq. ft. of main fuel bed per hour.....lb.

INDICATOR DIAGRAMS

(24) Mean effective pressure in lb. per sq. in.....lb.

SPEED AND EXPLOSIONS

(25) Revolutions per minute.....rev.
(26) Number of explosions per minute.....

POWER

(27) Indicated horsepower developed by engine.....i.h.p.

ECONOMY RESULTS

(28) Dry coal consumed per i.h.p. per hour.....lb.
(29) Cost of coal per i.h.p. per hour.....dollars
(30) Heat units consumed per i.h.p. per hour (Line 20 × Line 28)....B.t.u.

NOTE: For an engine driving an electric generator, the form may be enlarged to include electrical data in the manner given in the code for Complete Steam Power Plants, page 1755.

PART XV

WATERWHEEL CODE

1815

RULES FOR CONDUCTING TESTS OF WATERWHEELS

1 INTRODUCTION

Waterwheel tests may be divided into two classes, one of which may be termed "shop" tests and the other "field" tests. The former refer to those which are conducted in a plant devoted exclusively to testing work, and the latter to tests of the wheel in its permanent location. The Holyoke Water Power Company's testing flume is an example of a shop-testing plant, being one which is equipped for turbine water wheels of any size up to 300 h.p. at 18-ft. head. This plant, it is understood, is also the only one of the kind in the country which is available for commercial work. Under these circumstances there seems to be no call at the present time (1912) for a general code of rules applying to tests of that character. The tests to which the following code refers are therefore limited to field tests; the wheel being in place, and operating so far as possible under the conditions of service for which it was installed.

2 OBJECT AND PREPARATIONS

The usual object of a waterwheel test in the field is the determination of the capacity and efficiency of the wheel at various gate openings and if practicable at various speeds, as compared with standard or guaranteed performance. Having determined the object, whatever it may be, take the dimensions, note the physical condition of the wheel and of the plant throughout, install the testing appliances, etc., following the general instructions given on pages 1689, 1690, 1691, 1702 and 1703, so far as they pertain to the work in hand, and make preparations for the test accordingly.

The most important preparations are those which relate to the determination of the power developed by the wheel, and the quantity of water which it consumes. The nature of these preparations is governed altogether by the character of the equipment. As regards power determination, the simplest method is the one applying to a case where the wheel drives an electric generator and the power is

measured by calculation from the electrical output. Another simple method is one which may be used where the wheel serves as an auxiliary to steam power, and the output is determined by ascertaining the difference between the indicated horsepower developed by the engine when the wheel is on and that developed when the wheel is off. Another method which is applicable to almost any situation where there is room, although the most difficult of the three, is the use of a friction brake attached to the waterwheel shaft, being arranged so as to take the place of a section of the shaft which may temporarily be removed. As to preparation for water measurement, the desirability of preserving the maximum head of water usually makes it necessary to gage the stream supplied to the wheel or the stream leaving it, and to select or prepare for this purpose a sufficient length of canal having a uniform cross-section to determine the required velocity by float measurement. Another method consists in the use of current meters or pitometers which have been properly calibrated. In cases where some part of the head may be sacrificed either in the head race or tail race, the measurement may be made by the insertion of a suitable weir.

3 APPARATUS AND INSTRUMENTS

The apparatus and instruments required for a capacity and efficiency test of a waterwheel are:

(*a*) A friction brake, steam engine indicators, or electrical instruments, depending on the character of the equipment

(*b*) Graduated scales showing the heights of water in the flume above the wheel and in the discharge pit beneath

(*c*) One or more current meters or other apparatus for ascertaining the velocity of the water; or a weir

Directions for the use of these appliances may be found on pages 1699 to 1701 and in Appendices Nos. 18, 19, 21, and 5.

It is of the greatest importance that the water measured is that which is consumed wholly by the wheel. If water leaks by without going through the wheel, the quantity of leakage should be determined by independent measurement when the wheel is entirely shut off; in which case the gross quantity is corrected accordingly.

4 DURATION

The duration of a simple efficiency test of a waterwheel depends mainly upon the method of water measurement employed, and the

time required to obtain a sufficient number of observations to insure a reliable average. After the desired load and other conditions have been obtained, continuous observations and measurements for a period of 15 minutes is sufficient for all practical purposes, provided the water is measured by a weir, but a longer time is necessary when other methods of measurement are used.

5 RECORDS

The records should be obtained in a manner conforming to the principles explained on page 1703. Readings of the weight on the brake arm, levels of water in the flume and discharge pit, indications of the current meters, and revolutions per minute, should be taken every five minutes, and at more frequent intervals if they show much fluctuation. In case of float measurement, repeated observations should be made one after the other throughout the whole period of the trial.

6 CALCULATION OF RESULTS

The total average head of water on the wheel is obtained by adding together the reading of the scale in the flume and the vertical distance between the zero of this scale and that of the scale in the discharge pit, and subtracting the reading of the latter scale, both readings being taken in reasonably still water.

The velocity of water in the measuring canal is found by averaging the readings obtained at several points extending over the whole width of the canal.

The cubic feet of water consumed per minute is obtained by multiplying the cross-section of the stream in square feet by the velocity of the water in feet per minute, determined as stated above.

The total power of water available is obtained by multiplying the net weight of water in pounds consumed per minute by the total average head in feet on the wheel, and dividing the product by 33,000.

The brake horsepower delivered by the wheel is found by multiplying the net weight on the brake arm in pounds by the circumference of the corresponding circle in feet and by the number of revolutions per minute, and dividing the final product by 33,000.

7 DATA AND RESULTS

The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for or omitting those not required, as may conform with the object in view:

DATA AND RESULTS OF WATERWHEEL TEST ADAPTED TO BRAKE MEASUREMENT OF POWER

CODE OF 1912

(1) Test of.....water wheel located at.....
to determine....., conducted by.....
(2) Type of wheel and class of service.....
(3) Type of generator, if any, kind of current, etc.....
(4) Rated power of wheel.....h.p.
(5) Cross-section of stream where velocity of water is measured...sq. ft.

GENERAL DATA

(6) Date .....
(7) Duration of period covered by test.....hr.
(8) Average net weight on brake arm.....lb.
(9) Average revolutions per minute.....r.p.m.
(10) Total average head of water on wheel.....ft.
(11) Average velocity of water per minute in measuring canal.....ft.
(12) Cu. ft. of water consumed per minute.....cu. ft.
(13) Weight of water consumed per minute (Line 12 × 62.35).....lb.
(14) Leakage per minute.....lb.
(15) Net water consumed per minute (Line 13 — Line 15).....lb.

POWER

(16) Total power of water available.....h.p.
(17) Brake horsepower delivered by wheel.....br. h.p.

EFFICIENCY

(18) Efficiency of wheel (Line 17 ÷ Line 16).....per cent

THIRD SECTION

PART XVI

APPENDICES

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APPENDICES

APPENDIX No. 1. CLEARANCE MEASUREMENT.
APPENDIX No. 2. LEAKAGE TESTS OF ENGINES.
APPENDIX No. 3. GENERAL PRECAUTIONS REGARDING LEAKAGE.
APPENDIX No. 4. CALIBRATING WATER METERS.
APPENDIX No. 5. WEIRS, VENTURI METERS, ETC.
APPENDIX No. 6. PITOT TUBES.
APPENDIX No. 7. BITUMINOUS COAL SIZES.
APPENDIX No. 8. APPARATUS FOR PUMPING ENGINES.
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APPENDIX No. 34. LOCATION OF INSTRUMENTS FOR LOCOMOTIVE TESTS.

APPENDICES

APPENDIX NO. 1

CLEARANCE MEASUREMENT BY WATER

To measure the clearance by actual test, the engine is carefully set on the center, with the piston at the end where the measurement is to be taken. Assuming, for example, a Corliss engine, the best method is to remove the steam valve so as to have access to the whole steam port, and then fill up the clearance space with water, which is poured into the open port through a funnel. The water is drawn from a receptacle containing a quantity previously measured. When the whole space, including the port, is completely filled, the quantity left is measured, and the difference shows the amount which has been poured in. The measurement can be easily made by weighing the water, and the corresponding volume determined by calculation, making proper allowance for its temperature. The proportion required is the volume in cu. in. thus found, divided by the volume of the piston displacement, also in cu. in., and the result expressed as a decimal. In this test care should be taken that no air is retained in the clearance space when it is filled with water.

The only difficulty in measuring the clearance in this way is that occurring when the exhaust valves and piston are not tight, and the water poured in, flows away and is lost. If the leakage is serious, no satisfactory measurement can be made, and it is better to depend upon the volume calculated from the drawing. If not too serious, however, an allowance can be made by observing the length of time consumed in pouring in the water; then, after a portion of the water has leaked out, fill up the space again, taking the time and measuring the quantity thus added, determining in this way the rate of leakage. Data will thus be obtained for the desired correction.

APPENDIX NO. 2

LEAKAGE TESTS OF ENGINES, INCLUDING TIME METHOD

The method of testing the valves and pistons for leakage in a Corliss engine, or one in which the admission valves can be operated independently of the exhaust valves, is as follows:

First close the two steam valves, open the two indicator cocks, and admit a full pressure of steam into the chest by opening the throttle valve. The movement of the starting bar, first one way and then the other, so as to close one exhaust valve and then the other, causes the leakage through

the steam valves to escape from the open indicator cock, where it becomes visible. The quantity of leakage is judged by the force of the current of steam blowing out.

To test the exhaust valves and piston, block the flywheel so that the piston will be at a short distance from the end of the stroke, and turn on the steam. The leakage escapes to the exhaust pipe, and can be observed at the open atmospheric outlet. If the outlet is not visible, and there is a valve in the exhaust pipe, this can be shut and the indicator cock opened, thereby deflecting the steam which leaks, and causing it to appear at the indicator cock. In a condensing engine where no atmospheric pipe is provided, and there is no opening that can be made in the exhaust pipe in front of the condenser, some idea can be obtained in regard to the amount of leakage by observing how rapidly the condenser is heated.

It is well to make these tests with the piston in different positions, so as to cover the whole range of the length of the stroke.

Another method of testing leakage is called the "time method." Instead of observing the steam that actually blows through the valves or piston to be tested, they are subjected to full steam pressure, and when the parts are thoroughly heated, the throttle valve is shut and the length of time observed which is required for the pressure to disappear. In testing the piston and exhaust valves, the flywheel is blocked as before, and, preferably, as indicator is attached, and a line drawn on a blank card at intervals of, say, one-quarter of a minute after the valve is shut, thereby making a record of the fall of the pressure. In a tight engine the fall of the pressure is slow, whereas in a leaky engine it is sometimes very rapid. The relative condition of the engine as compared with a tight engine must be judged by an observer, who must, of course, have had experience in tests of this kind on engines in various conditions.

The leakage of a piston can always be determined by removing the cylinder head and observing what blows through the open end with the pressure of steam behind it. The advantage of the time method is that it saves the labor and time required in removing the cylinder head and replacing it, which, in cases of large engines, is considerable.

Leakage tests of single-valve engines cannot be made as satisfactorily as those of the Corliss type and other four-valve engines. The best that can be done as regards the valve is to place it at or near the center of its travel, covering both ports, and then make the test under full pressure. The valve and piston can be tested as a whole by blocking the flywheel and opening the throttle valve in the same way as in other engines. In locomotives leakage will be revealed by the escape of steam at the top of the smoke stack.

In testing compound engines for leakage, the work is somewhat simplified in case of any one cylinder, as compared with a single engine. For example, leakage of the high-pressure cylinder can be revealed by opening the indicator cock on the proper end of the low-pressure cylinder, the steam valve of that cylinder being open. The test of leakage of the low-pressure exhaust valves and piston when the time method is used can be based on the indications of the receiver gage, instead of using an indicator. In that case the fall of the pressure due to leakage is read directly from the gage.

The tests thus far referred to are qualitative, and not quantitative. It is practical in some cases to determine the quantity of leakage under any set of conditions by collecting the steam which passes through, condensing it and weighing it. This can be readily done when there is a surface con-

denser, and it can be done in the absence of such a condenser by attaching a small pipe to the exhaust, and carrying the steam which escapes into a tank of water and condensing it. How much dependence can be placed upon the results of such a quantitative test as showing the actual quantity of leakage which occurs when the valves and pistons are in motion must be left to the judgment of the person who makes the test.

In Corliss engines the leakage of the piston with the engine in operation can be observed by removing the cylinder head, disconnecting the steam and exhaust valves at the head end, and setting the engine to work with steam admitted at the crank end.

APPENDIX NO. 3

GENERAL PRECAUTIONS REGARDING LEAKAGE AND METHODS OF MEASURING LEAKAGE

(*a*) *Leakage*

It is not always necessary to blank off a connecting pipe to make sure that there is no leakage through it. If satisfactory assurance can be had that there is no chance for leakage, this is sufficient. For example, where a straightway valve is used for cutting off a connecting pipe, and this valve has double seats with a hole in the bottom between them, this being provided with a plug or pet cock, assurance of the tightness of the valve when closed can be had by removing the plug or opening the cock. Likewise, if there is an open drip pipe attached to an unused or empty section of pipe beyond the valve, the fact that no water escapes here is sufficient evidence of the tightness of the valve. The main thing is to have positive evidence in regard to the tightness of the connections, such as may be obtained by the means suggested above; but where no positive evidence can be obtained, or where the leakage that occurs cannot be measured, it is of the utmost importance that the connections should be broken and blanked off.

Leakage of relief valves which are not tight, drips from traps, separators, etc., and leakage of tubes in the feedwater heater must all be guarded against or measured and allowed for.

It is well, as an additional precaution, to test the tightness of the feedwater pipes and apparatus concerned in the measurement of the water by running the pump at a slow speed for, say, fifteen minutes, having first shut the feed valves at the boilers and making sure they are tight. Leakage will be revealed by disappearance of water from the supply tank. In making this test, a gage should be placed on the pump discharge to guard against undue or dangerous pressure.

(*b*) *Water Glass Tests of Leakage*

To determine the leakage of steam and water from a boiler and steam pipes, etc., the water-glass method may be satisfactorily employed. This consists of shutting off all the feed valves (which must be known to be tight) and the main feed valve, thereby stopping absolutely the entrance or exit of water at the feed pipes to the boiler; then maintaining the steam pressure (by means of a very slow fire) at a fixed point, which is approxi-

nately that of the working pressure, and observing the rate at which the water falls in the gage glasses. It is well, in this test, as in other work of this character, to make observations every ten minutes, and to continue them for such length of time that the differences between successive readings attain a constant rate. In many cases the conditions will have become constant at the expiration of fifteen minutes from the time of shutting the valves, and thereafter the fall of water due to leakage of steam and water become approximately constant. It is usually sufficient, after this time, to continue the test for two hours, thereby obtaining a number of half-hourly periods. When this test is finished, the quantity of leakage is ascertained by calculating the volume of water which has disappeared, using the area of the water level and the depth shown on the glass, making due allowance for the weight of one cubic foot of water at the observed pressure. The water columns should not be blown down during the time a water-glass test is going on, nor for a period of at least one hour before it begins.

If there is opportunity for condensation to occur and collect in the steam pipe during the leakage test, the quantity should be determined as closely as desirable, and properly allowed for.

(*c*) *Surface Condenser Tests*

In making an engine test where the steam consumption is determined from the amount of water discharged from a surface condenser, leakage of the piston rods and valve rods should be guarded against; for if these are excessive, the test is of little use, as the leakage consists partly of steam that has already done work in the cylinder and of water condensed from the steam when in contact with the cylinder. If such leakage cannot be prevented, some allowance should be made for the quantity thus lost. The weight of water as shown at the condenser must be increased by the quantity allowed for this leakage.

APPENDIX NO. 4

CALIBRATING WATER METERS

Referring to Fig. 2, two tees *A* and *B* are placed in the feed pipe, and between them two valves *C* and *D*. The meter is connected between the outlets of the tees *A* and *B* and the valves *E* and *F* are placed one on each side of the meter. When the meter is running, the valves *E* and *F* are opened, and the valves *C* and *D* closed. A small bleeder *G* is kept open to make sure that there is no leakage. A gage is attached at *H*. When the meter is tested, the valves *C*, *D*, and *F* are closed, and the valves *E* and *I* are opened. The water flows from the valve *I* to a tank on platform scales. In testing the meter, the water is throttled at the valve *I* to obtain the desired rate of discharge, the gage meanwhile showing the working pressure. The piping leading from the valve *I* to the tank is arranged with a swinging joint, consisting merely of a loosely fitting elbow, so that it can be readily turned into the tank or away from it. When the desired speed has been secured, the end of the pipe is swung into the tank at the instant the pointer

of the meter is opposite some graduation mark on the dial. When the required number of cubic feet are discharged, the pipe is swung away. The tests should start and stop at the same graduation mark on the first dial, and continued until at least 10 or 20 cu. ft. are discharged for one test. The tank is weighed before and after filling.

The water passing the meter should always be under pressure so that any air in the meter may be discharged through the vents provided for this pur-

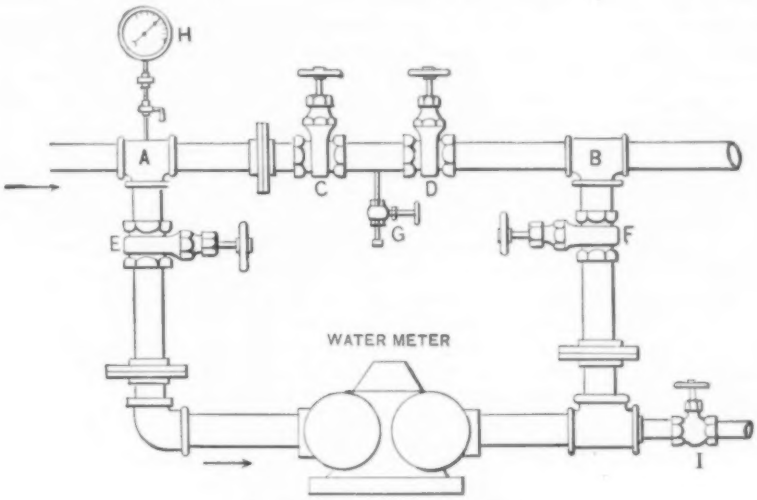


FIG. 2 METER CALIBRATION

pose. Care should be taken that there is no unnecessary air drawn into the feed water. The meter should be tested before and after the trial, and repeated calibrations should be made to obtain confirmative results.

Fig. 2* and the description apply to a piston meter, but any other type of meter carrying water under pressure may be calibrated in the same manner.

APPENDIX NO. 5

MEASUREMENTS OF WATER BY MEANS OF WEIRS, VENTURI METERS, ETC.

Weirs

The measurement of water by the use of weirs may be based on Francis' experiments, an account of which is given in Lowell's Hydraulic Experiments.[1] These resulted in the formula

$$Q = 3.33\ (L - 0.2H) \times H^{\frac{3}{2}}$$

*Reproduced from Trans. Am. Soc. M. E., vol. 24, p. 724, fig. 118.

[1] D. Van Nostrand.

in which Q is the discharge in cu. ft. per second, L the length of the weir in ft., and H the depth of water on the weir in ft. The coefficient, 3.33 was obtained from the mean of 88 experiments, the greatest variation from the mean in any individual case being 1 per cent. The length of the weir in all but six of the experiments, was approximately 10 ft. The depth of water on the weir varied from 7 to 19 in. The formula applies to that type of weir having perfect contraction at each end, which was the form used in 65 experiments.

The weir was of rectangular cross-section, with a horizontal crest and vertical ends. The upper edge was made of cast iron, and the corner presented to the current was square and sharp. The horizontal part of the crest was ¼ in. wide, and the remaining part was bevelled off at an angle of 45 deg. The ends were of similar cross-section to the crest. The depth on the weir was taken by means of hook gages, 6 ft. from the weir, these gages being placed in wooden boxes situated on the sides of the canal and communicating with the water through small openings. Vertical gratings, for overcoming eddies in the current, were provided above the gages. The distance from the side of the canal to the end of the weir was about 2 ft., and the depth of the canal below the crest was, in most of the experiments, about 5 ft.

The Francis formula is applicable only to cases similar to those described. According to Mr. Francis' statement, it cannot be applied where the depth on the weir exceeds one-third of the length, nor to very small depths. Furthermore, the distance from the side of the canal to the end of the weir should not be less than three times the depth on the weir.

In using the formula, the depth should be corrected for the head due to the velocity of approach, according to the formula

$$H' = \left[\left(H + h\right)^{\frac{3}{2}} - h^{\frac{3}{2}}\right]^{\frac{2}{3}}$$

in which H' is the corrected depth, H the observed depth, and h the head due to the velocity of approach. This last may be determined from the formula

$$h = \frac{V}{64.3}$$

in which V is the velocity of approach in ft. per second, which may be determined by dividing the uncorrected discharge of water, in cu. ft. per second, by the area of the cross-section of the steam flowing through the canal, in sq. ft.

Refer also to the experiments on weirs made by Hamilton Smith, Jr., described in Smith's Hydraulics, and to those made by Fteley and Stearns, described in the Transactions of the American Society of Civil Engineers, 1883.

Venturi Tubes

According to Herschel's experiments, described in the Transactions of the American Society of Civil Engineers, November 1887 and January 1888, a venturi tube inserted in a force main may be used for determining the quantity of water discharged by a pumping engine. Such a tube, applied, for example, to a 24-in. main, has a total length of about 20 ft. At a distance of 4 ft. from the end nearest the engine, the inside diameter of the tube is contracted to a throat having a diameter of about 8 in. A pressure

gage is attached to each of two chambers, the one surrounding and communicating with the entrance, or main pipe, the other with the throat. Experiments made upon two tubes of this kind, one of which was 4 in. in diameter at the throat and 12 in. at its entrance, and the other about 36 in. in diameter at the throat and 9 ft. at its entrance, showed that the quantity of water which passes through the tube is very nearly the theoretical discharge through an opening having an area equal to that of the throat, and a velocity that due to the difference in head shown by the two gages. Mr. Herschel states that the coefficient for these two widely varying sizes of tubes and for a wide range of velocity through the pipe was found to be within 2 per cent, either way, of 98 per cent. In other words, the quantity of water flowing through the tube per second is expressed within 2 per cent by the formula

$$W = 0.98 \times A \times \sqrt{2gh}$$

in which A is the area of the throat of the tube, and h the head, in ft., corresponding to the difference in the pressure of the water entering the tube and that found at the throat.

The substantial reliability of venturi meters of small size (6 in. at entrance and 2 in. at throat), such as are readily calibrated, furnishes strong evidence of the practicability of this method of measurement for pumping engines.

Nozzles

The measurement of water by computation from its discharge through the nozzles of fire hose, furnishes a means of determining the quantity of water delivered by a pumping engine which can be applied without much difficulty. Freeman's investigations upon fire nozzles, described in the Transactions of the American Society Civil Engineers, November, 1889, which covered a wide range of pressures and sizes, showed that the coefficient of discharge for a smooth nozzle of ordinary good form was within ½ of 1 per cent, either way, of 0.977; the diameter of the nozzle being accurately calipered, and the pressures being determined by means of an accurate gage attached to a suitable piezometer at the base of the play pipe.

To use this method the water should be conducted to a pressure box and as many nozzles attached to the box as would be required to carry off the water. Four 1¼-in. nozzles, with a pressure of 80 lb. per sq. in., would discharge the full capacity of a 2,500,000 gal. engine.

Orifice Formula

The quantity of water flowing through an orifice into the open air, expressed in cu. ft. per second, may be calculated by the use of the formula

$$8.02\ AC\sqrt{H}$$

in which A is the area of the orifice in sq. ft., C, a coefficient depending on the form of the orifice and velocity of approach, and H, the head over the center of the orifice measured in ft. For an orifice located in the side of a tank, and consisting of a circular opening in a thin metal plate with a smooth sharp edge, the value of the coefficient is very close to 0.6. (See Hamilton Smith, Jr.'s, work on Hydraulics.)

Float Rods and Current Meters

The velocity of water flowing in a stream may be obtained indirectly by the use of float rods, and directly by means of current meters.

Float rods may be patterned after those used in Francis's experiments, which consisted of tin tubes, 2 in. in diameter, the lower ends being loaded so as to float upright, and the upper ends standing a few inches out of water. They were used at each foot of width of the canal, the length of run being 70 ft., and the time was recorded on a chronograph. The mean velocity ranged from 0.5 ft. to 5 ft. per second, the number of experiments being 115. It was found that if the lower ends of the rods did not extend to near the bottom of the canal, the observed velocity should be multiplied by a coefficient obtained from the formula

$$1 - 0.116\ (\sqrt{D} - 0.1)$$

in which D is the proportionate part of the total depth of the water not reached by the float. If for example the immersed part of the float is 0.9 of the total depth, D is 0.1 and the coefficient becomes $1 - 0.024 = 0.976$.

A current meter consists essentially of a small screw propeller which is operated by the current of flowing water in which it is immersed, the revolutions being shown on a dial placed at some convenient point to which the motion is transmitted electrically. The speed of revolution of the propeller, referred to a rating table, shows the velocity of the water. The Price electric meter is an example of a satisfactory instrument for this purpose. Whatever type of meter is used, its rating should be checked by calibration under conditions which are substantially the same as those occurring on the waterwheel test, especially those pertaining to depth of immersion and average velocity. The method of calibration considered most reliable is that which is carried on by moving the meter through still water at a known velocity.

For further information regarding these and other methods of measuring the velocity of water, reference may be made to Water Supply and Irrigation Paper No. 95, U. S. Geological Survey, published in 1904, on the subject, Accuracy of Stream Measurements.

APPENDIX NO. 6

PITOT TUBES FOR GAS AND AIR MEASUREMENT

A form of pitot tube suitable for measuring the velocity of gas or air moving in a pipe is illustrated in principle in Fig. 3. This consists of a small tube *A*, which is inserted in such a position that the open end points toward the current of gas or air, thereby receiving the full impact which the current produces. Surrounding this tube is an annular chamber or jacket *B*, which communicates with the outside space by means of four small holes *C*, *D*, *E*, and *F*, two on each side. By means of the small pipes *G* and *H*, the inner tube and the outer jacket are connected respectively to the two legs of a U-tube manometer, as shown in Fig. 4.

The pressure shown by the manometer, or what is called the velocity-

pressure, is that due to the difference in pressure at the two points, the inner tube *A* being subjected to the sum of the static and velocity pressures, and the annular jacket *B* to the static pressure alone. The velocity corresponding to the difference of pressure shown is computed by using the formula

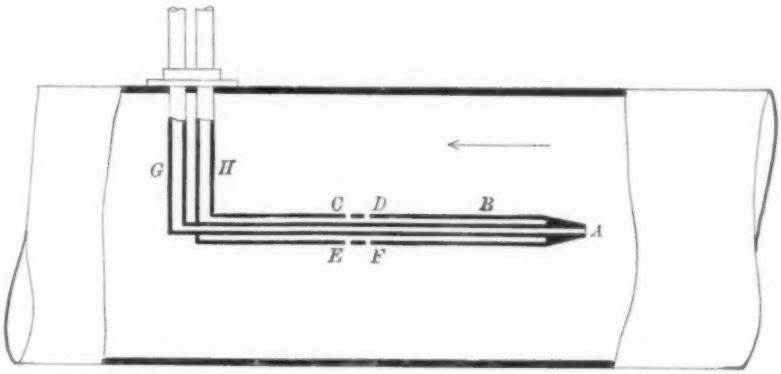


FIG. 3 PITOT TUBE

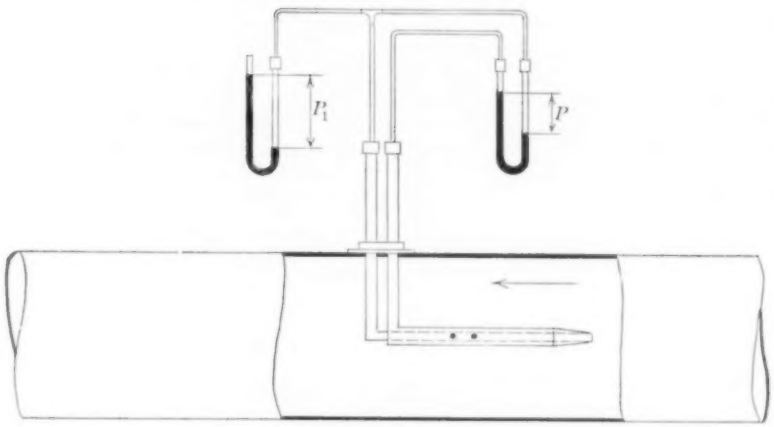


FIG. 4 PITOT TUBE AS INSTALLED

$$\text{Velocity per minute} = 60\sqrt{2gH}$$

in which $H = \dfrac{\text{Velocity-presssure in lb. per sq. ft.}}{\text{Weight of 1 cu. ft. of gas or air.}}$

The velocity-pressure in lb. per sq. ft. is found by multiplying the indication of the manometer *P* in in. of water by the constant 5.196. By substituting in the above formula, and reducing, we have

$$\text{Velocity per minute} = 1097\sqrt{\frac{\text{Reading of manometer in in. of water.}}{\text{Weight of 1 cu. ft. of gas or air.}}}$$

The weight of 1 cu. ft. of gas or air is obtained by multiplying the weight

of 1 cu. ft. at a temperature of 32 deg. and pressure of 29.92 in. (0.0807 lb. for air) by the proportion obtained from the expression

$$\frac{492 \text{ deg.}}{460 + t} \times \frac{P + 29.92}{29.92}$$

in which t is the temperature of the gas or air as observed and P the observed pressure in in. of mercury.

For low velocities the U-tube shown in Fig. 4 may be replaced by one containing gasolene and having one leg inclined so as to multiply the ordinary indications to any extent desired. In this case the reading may be converted into inches of water by multiplying it by the proportion of the vertical height to the inclined height and by the specific gravity of gasolene.

For high pressures, the manometer attached to the impact tube can be replaced by a mercury column or steam gage attached to a receiver in the main where the velocity is largely reduced.

To determine the average velocity of the gas or air throughout the whole cross-section of the pipe, readings should be obtained at a number of points and the results averaged. Two sets of readings should be taken, one set along the horizontal diameter and the other set on the vertical diameter. The various points should be selected so that the different velocities will conform to equal areas. If there are ten points in each diameter this equality will be obtained if they are located on the four radii at distances of 32, 55, 71, 84, and 95 per cent, respectively, of their length, measured from the center. In pipes carrying high pressures two pitot tubes may be used, one for each diameter, and they should be mounted so as to be readily moved to the desired locations, the required points being indicated by suitable marks on the outside supports.

The point selected for the attachment of a pitot tube should be as far removed as practicable from abrupt bends.

Another form of tube may be used, when properly calibrated, especially for high pressures, which is inserted diametrically across the main in the manner of a sampling pipe. In this case the impact is obtained by a number of short tubes attached to the side and pointing toward the current.

APPENDIX NO. 7

BITUMINOUS COAL SIZES

Bituminous coals in the Eastern States may be graded and sized as follows:

(*A*) Run of mine coal; the unscreened coal taken from the mine.

(*B*) Lump coal; that which passes over a bar-screen with openings 1¼-in. wide.

(*C*) Nut coal; that which passes through a bar-screen with 1¼-in. openings and over one with ¾-in. openings.

(*D*) Slack coal; that which passes through a bar-screen with ¾-in. openings.

Bituminous coals in the Western States may be graded and sized as follows:

(*E*) Run of mine coal; the unscreened coal taken from the mine.

(*F*) Lump coal; divided into 6-in., 3-in. and 1¼ in. lump, according to the diameter of the circular openings over which the respective grades pass; also 6 by 3 lump and 3 by 1¼ lump, according as the coal

passes through a circular opening having the diameter of the larger figure and over that of the smaller diameter.

(*G*) Nut coal; divided into 3-in. steam nut, which passes through an opening 3-in. diameter and over 1¼-in.; 1¼-in. nut, which passes through a 1¼-in. diameter opening and over a ¾-in. diameter opening; ¾-in. nut, which passes through a ¾-in. diameter opening and over a ⅝-in. diameter opening.

(*H*) Screenings; that which passes through a 1¼-in. diameter opening.

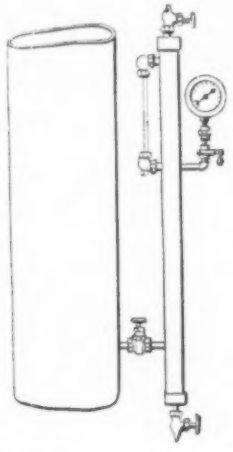


FIG. 5 AIR CHAMBER FOR ATTACHMENT OF GAGE TO FORCE MAIN

(*I*) Washed sizes; those passing through or over the circular openings of the following diameters, in inches:

Number	Through	Over
1	3	1¾
2	1¾	1⅛
3	1⅛	¾
4	¾	¼
5	¼	...

APPENDIX NO. 8

APPARATUS FOR PUMPING ENGINES

To determine the length of stroke in the case of direct-acting engines, a scale should be securely fastened to the frame which connects the steam and water cylinders, in a position parallel to the piston rod, and a pointer attached to the rod so as to move back and forth over the graduations on the scale. The marks on the scale, which the pointer reaches at the two ends of the stroke, are thus readily observed, and the distance moved over computed. If the length of the stroke can be determined by the use of some form of registering apparatus, such a method of measurement

is preferred. The personal errors in observing the exact scale marks, which are liable to creep in, may thereby be avoided.

The apparatus referred to under the heading (*g*) *Pressure Gages*, page 1694, consisting of a small air chamber for the attachment of a gage to the force main is shown in Fig. 5*.

APPENDIX NO. 9

HUMIDITY TABLE

Table 6 is condensed from the hygrometric tables of U. S. Government referred to on Page 1696.

APPENDIX NO. 10

OPTICAL, PNEUMATIC, AND RADIATION PYROMETERS

Wanner's optical pyrometer, Uehling's pneumatic pyrometer, and Ferry's radiation pyrometer, are among those referred to on page 1696 as being useful for measuring high temperatures. Full descriptions of these instruments may be found in the catalogues of various dealers in chemical and physical apparatus.

APPENDIX NO. 11

DESCRIPTION OF STEAM CALORIMETERS

The instrument referred to on page 1697 is shown in Figs. 6*a* and 6*b*. It consists of brass pipe and fittings of the ½-in. size in which the leading parts are the two tees, a pair of flanges, an orifice plate between the flanges, and two thermometer cups containing thermometers which are screwed into the tees. The whole is surrounded by suitable non-conducting covering and encased in a box. The percentage of moisture is determined by observing the number of degrees of cooling that the thermometer in the low-pressure steam shows the "normal" reading for dry steam, and dividing that number by the "constant" number of degrees representing 1 per cent of moisture.

To determine the "normal" reading of the low-pressure thermometer corresponding to dry steam, the instrument should be attached to a horizontal steam pipe in such a way that the sampling nozzle projects upwards to near the top of the pipe, there being no perforations and the steam entering through the open top of the nozzle. The test should be made when the steam in the pipe is in a quiescent state, and when the steam pressure is maintained constantly at the point observed on the main trial. If the steam pressure falls during the time when the observations are being made,

* Reproduced from Trans. Am. Soc. M. E., vol. 12, p. 548, fig. 109.

the test should be continued long enough to obtain the effect of an equivalent rise of pressure.

To find the "constant" for 1 per cent of moisture divide the latent heat of the steam supplied to the calorimeter at the observed pressure or tempera-

TABLE 6. RELATIVE HUMIDITY, PER CENT.

Dry Thermometer, Deg. Fahr.	Difference between the Dry and Wet Thermometers, Deg. Fahr.													
	1	2	3	4	5	6	7	8	9	10	11	12	13	14
	Relative Humidity, Saturation being 100 (Barometer = 30 in.)													
32	89	79	69	59	49	39	30	20	11	2	..	..	..	..
40	92	83	75	68	60	52	45	37	29	23	15	7	0	..
50	93	87	80	74	67	61	55	49	43	38	32	27	21	16
60	94	89	83	78	73	68	63	58	53	48	43	39	34	30
70	95	90	86	81	77	72	68	64	59	55	51	48	44	40
80	96	91	87	83	79	75	72	68	64	61	57	54	50	47
90	96	92	89	85	81	78	74	71	68	65	61	58	55	52
100	96	93	89	86	83	80	77	73	70	68	65	62	59	56
110	97	93	90	87	84	81	78	75	73	70	67	65	62	60
120	97	94	91	88	85	82	80	77	74	72	69	67	65	62
140	97	95	92	89	87	84	82	79	77	75	73	70	68	66

	15	16	17	18	19	20	21	22	23	24	26	28	30
32	..	..	..	..	..	..	..	..	..	..	..	..	..
40	..	..	..	..	..	..	..	..	..	..	..	..	..
50	11	5	0	..	..	..	..	..	..	..	..	..	..
60	26	21	17	13	9	5	1	..	..	..	..	..	..
70	36	33	29	25	22	19	15	12	9	6	..	..	..
80	44	41	38	35	29	26	23	20	18	12	7	..	..
90	49	47	44	41	39	36	34	31	29	26	22	17	13
100	54	51	49	46	44	41	39	37	35	33	28	24	21
110	57	55	52	50	48	46	44	42	40	38	34	30	26
120	60	58	55	53	51	49	47	45	43	41	38	34	31
140	64	62	60	58	56	54	53	51	49	47	44	41	38

ture by the specific heat of superheated steam at atmospheric pressure (0.46) and divide the quotient by 100.

Finally ascertain the percentage of moisture by dividing the number of degrees of cooling by the constant, as above noted.

To determine the quantity of steam used by the calorimeter it is usually sufficient to calculate the quantity from the area of the orifice and the absolute pressure, using Napier's formula for the number of lb. which passes through per second; that is, absolute pressure in lb. per sq. in. divided by 70 and multiplied by the area of orifice in sq. in. To determine the quan-

tity by actual test, a steam hose may be attached to the outlet of the calorimeter, and carried to a barrel of water on platform scales. The

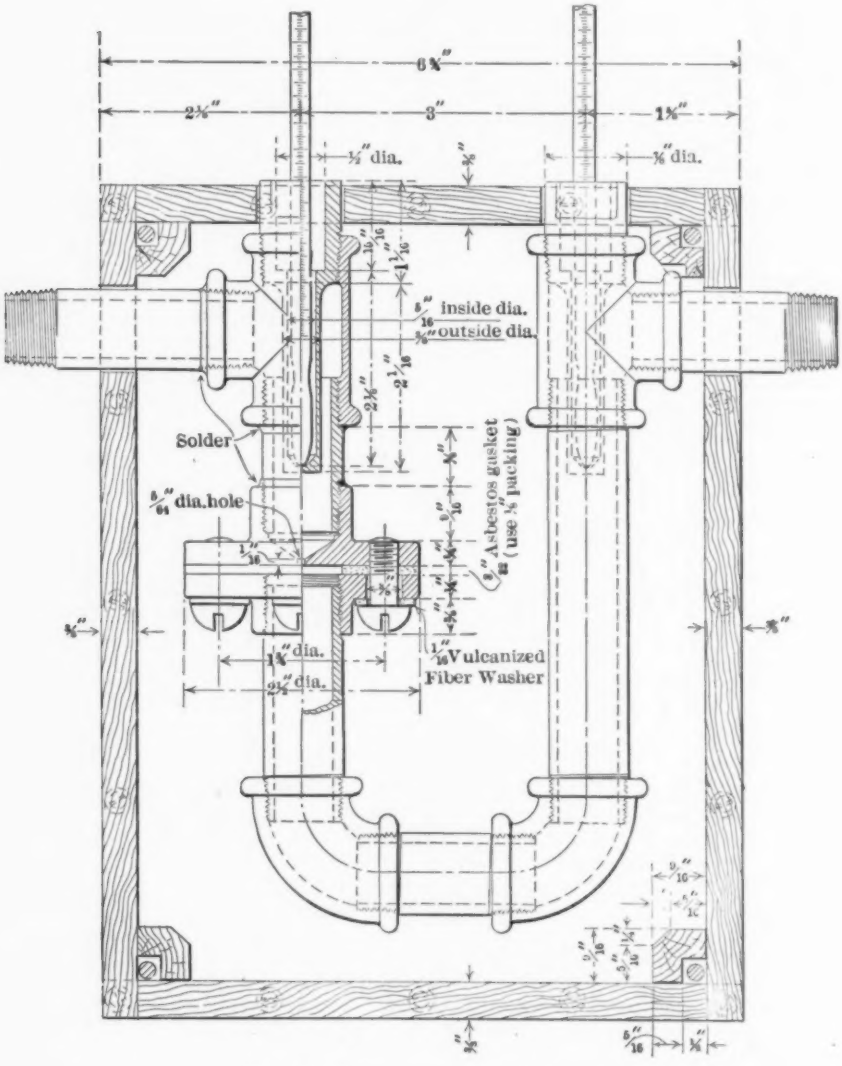


FIG. 6*a* THROTTLING CALORIMETER

amount of steam condensed in a certain time is determined, and thereby the quantity discharged per hour.

A combined throttling and separating calorimeter, in which the throt-

tling portion operates on the same principle as the one just described, is shown in Volume 17 of the Transactions.[1]

APPENDIX NO. 12

MAHLER COAL CALORIMETER

The Mahler coal calorimeter consists essentially of a strong cylindrical vessel having a capacity of about 800 cu. c., which is closed at the top and filled with oxygen gas, under a pressure of 375 lb. per sq. in. A sample of finely powdered coal, which will pass through a sieve having 10,000 meshes to the sq. in., weighing about 1 gram, is placed in a pan suspended within the interior vessel, and provided with two electrodes, through which an electric current from a battery can be passed. The whole is immersed in

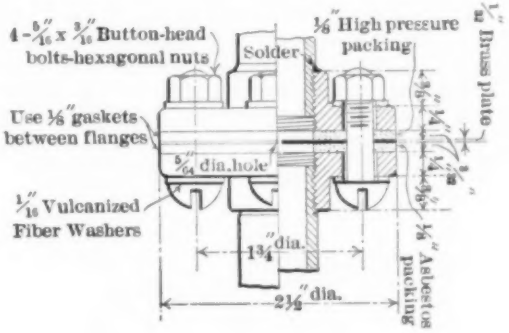


FIG. 6b ORIFICE PLATE FOR THROTTLING CALORIMETER

an outer vessel containing about 2500 grams of water, thoroughly stirred, the temperature of the water observed, the coal set on fire by completing the electric circuit, the water again stirred, and the temperature observed at intervals of half a minute until the thermometer ceases to rise. The difference between the initial and final temperatures thus determined is corrected for radiation, the latter being found by observing the rate at which the temperature changes before and after the coal is fired. The weight of water contained in the outer vessel, the water equivalent of the whole apparatus, and the weight of coal being known, the total heat of combustion is found by multiplying the sum of the two water weights in grams by the corrected rise of temperature in deg. fahr., and the product is divided by the weight of the coal in grams.

The sample used for the calorimeter test should be air dried. A similar air-dried sample should be tested for moisture in the manner described at the end of page 1712, the final result being based on dry coal.

[1] R. C. Carpenter. New Form of Steam Calorimeter. Trans. Am. Soc. M. E., vol. 17, p. 618.

APPENDIX NO. 13

JUNKER GAS CALORIMETER

This instrument consists of a vertical cylindrical water chamber containing vertical tubes, which is heated by the gas burned in a Bunsen lamp beneath. The products of combustion pass upward through the tubes, and the water flows downward through the chamber in a continuous current. The quantity of gas is measured by a gas meter, and the quantity of water by collecting the overflow discharged from the apparatus. Thermometers are inserted at the points of entrance and exit. The heat of combustion of a cu. ft. of gas is determined by multiplying the rise of temperature in deg. fahr. by the weight of water in lb., and dividing the product by the volume of gas in cu. ft. The result thus found is what is termed the "higher value," and this is the value, unless otherwise stated, which is employed throughout the codes.

The "low value" is obtained by multiplying the weight of the condensed vapor resulting from the combustion, expressed in lb., by the total heat of atmospheric steam above the temperature of the condensed vapor, dividing the product by the volume of the gas in cu. ft., and subtracting the quotient from the higher value.

APPENDIX NO. 14

FUEL ANALYSES

Proximate Analysis of Coal

The apparatus required for proximate analysis consists of a mill for grinding coal, chemical scales sensitive to 1/1000 of the amount weighed, drying apparatus, a platinum crucible, a Bunsen burner and blast lamp, a supply of oxygen gas, and such chemicals and chemical apparatus as may be required. The elements to be determined are moisture, volatile matter, fixed carbon, ash, and sulphur.

Determine the moisture as explained at the end of Part III *A*.

To determine volatile matter, place one gram of air-dried powdered coal in the crucible and cover it with a loose platinum plate. Heat 3½ minutes in the flame of the Bunsen burner, and continue the heating for 3½ minutes longer in the flame of the blast lamp. Cool down, remove the cover, and weigh the residue. The loss in weight represents the combined volatile matter and moisture. Subtracting the moisture, the weight of volatile matter alone is determined.

To ascertain the ash, expose the residue in the crucible to the blast lamp until it is completely burned, using a stream of oxygen if desired to hasten the process. The residue left is the ash.

The difference between the residue left after the expulsion of the volatile matter and the ash is the fixed carbon.

To determine sulphur by Eschka's method, which is the one commonly used, heat 1 gram of coal mixed with 1 gram of magnesium oxide and ½ gram of sodium carbonate for 1 hour, using an alcohol lamp. After cooling, mix with 1 gram of ammonium nitrate and heat the mixture ten minutes; then dissolve in 200 cc. of water, heat and reduce by evaporation to 150 cc., acidify with hydrochloric acid, and filter. Add barium chloride to the filtrate and determine the sulphur by calculation from the quantity and composition of the barium sulphate thereby precipitated.

Ultimate Analysis of Coal

The apparatus required for ultimate analysis consists of a mill and other apparatus for grinding and pulverizing the coal; chemical scales sensitive to 1/1000 of the amount weighed; drying apparatus; combustion apparatus, embracing a combustion furnace, a glass combustion tube, one end of which is filled with copper oxide and chromate of lead and the other end with a roll of oxidized copper gauze, a porcelain boat, a set of bulbs containing hydrate of potassium, a U-tube filled with chloride of calcium, and a supply of pure oxygen and pure air; together with suitable chemicals and chemical apparatus required for the various processes. The elements to be determined are moisture, carbon, hydrogen, oxygen, sulphur, nitrogen, and ash.

The moisture is determined in the manner pointed out under Proximate Analysis.

The carbon and hydrogen are obtained by the use of the combustion apparatus. One-half gram of the pulverized air-dried coal is placed in the porcelain boat, which is introduced between the copper roll and the copper oxide within the combustion tube. After the coal and the entire contents within have been thoroughly dried out by a sufficient preliminary heating, aided by a current of dry air, the furnace is set to work and the coal burned by first passing air through the tube and finally oxygen, conducting the products of combustion through the potash bulbs and the chloride of calcium tube. The carbon dioxide produced by the combustion of the carbon is absorbed by the potash, and the water formed by the combustion of hydrogen, together with that due to the moisture in the air-dried coal, is taken up by the chloride of calcium. The quantity of carbon is determined by weighing the bulbs before and after, thereby obtaining the weight of the carbon dioxide produced, and then calculating the weight of carbon from the known composition of the dioxide. Likewise, the quantity of hydrogen is determined by weighing the calcium tube before and after, which, after deducting the moisture in the air-dried coal, gives the amount of water produced, and, dividing by 9, the amount of hydrogen.

Sulphur is found by the method described above under the heading Proximate Analysis.

To determine nitrogen, a certain weight of coal is mixed with strong sulphuric acid and permanganate of potash and heated until nearly colorless. This process converts the nitrogen into ammonia and then into sulphate of ammonia, and the amount of sulphate is determined by making the

solution alkaline and then distilling it. The nitrogen is found by calculation from the known composition of ammonia.

The ash is found by weighing the refuse left in the combustion boat after the coal is completely burned.

The oxygen is the difference between the sum of the elements previously determined and the original weight of coal.

The ultimate analysis of coal, as will be seen from the above description, requires the use of so much chemical apparatus, and at best it is so complicated, that it is not likely to be done except in a fully equipped chemical laboratory. It should not be undertaken by one who is not entirely familiar with all the details of the work.

Analysis of Liquid Fuels

The determination of carbon and hydrogen in liquid fuels is made in the same manner as that concerning the solid fuels above described, using special means for preventing loss in the various processes on account of the volatile characteristics of the fuel.

To determine the sulphur, the oil or other liquid is heated with nitric acid and barium chloride. The quantity of sulphate of barium thus produced is ascertained by filtering and weighing, and the sulphur calculated from the known composition of the compound.

The ultimate analysis of liquid fuel, like that of coal, should be undertaken only by a person familiar with all the necessary details.

See report of Committee of American Chemical Society on Coal Sampling and Analysis.

APPENDIX NO. 15

GAS ANALYSIS

Orsat Apparatus

The Orsat apparatus is a portable instrument contained in a wooden case with removable sliding doors front and back, as shown in its simplest form in Fig. 7*. It consists essentially of a measuring tube or burette, three absorbing bottles or pipettes, and a leveling bottle, together with the connecting tubes and apparatus. The bottle and measuring tube contain pure water; the first pipette, sodium hydrate dissolved in three times its weight of water; the second, pyrogallic acid dissolved in sodium hydrate in the proportion of 5 grams of the acid to 100 cc. of the hydrate; and the third, cuprous chloride. These chemicals are sold by most of the large dealers.

The manipulation of the instrument, which can be carried on after suitable practice by any person familiar with testing work, is as follows:

After completely drawing out the air contained in the supply pipe, a sample of the gas is drawn into the measuring tube by opening the

* Reproduced from Trans. Am. Soc. M. E., vol. 18, p. 903, figs. 290, 291.

necessary connections and allowing the water to empty itself from the tube and flow into the bottle. The quantity of gas drawn in is adjusted to 100 cc. By opening one by one the connections to the pipettes, and raising and lowering the water bottle, the sample is alternately admitted to and withdrawn from the pipettes, and the ingredients one by one absorbed.

The first pipette absorbs carbon dioxide (CO_2); the second, oxygen (O); and the third, carbonic oxide (CO). The quantity absorbed in each case is determined by finally returning the sample to the measuring burette and reading the volume. The percentage of CO_2 is read directly, being the first absorption. That of the other two

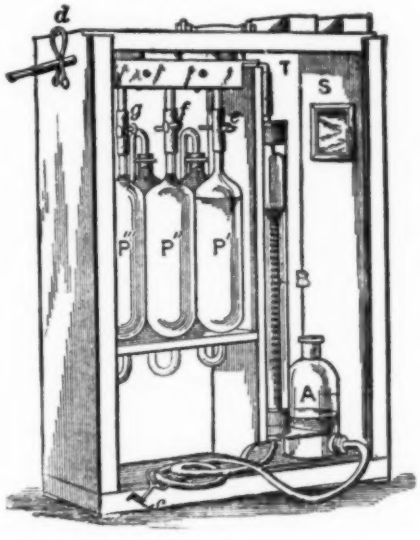


FIG. 7 ORSAT APPARATUS

ingredients are the respective differences between the readings taken after successive absorptions.

Various modifications of this apparatus have been developed which enable analyses to be made with greater rapidity than with the form illustrated in Fig. 7.

Hempel Apparatus

The Hempel apparatus works on the same principle as the simple form of Orsat apparatus described, so far as the latter is applicable, excepting that the absorption may be hastened by shaking the pipettes bodily, bringing the chemical into most intimate contact with the gas. It is less portable and in some particulars it requires more careful manipulation than the Orsat, while for general analysis it is not adapted unless used in a well

equipped chemical laboratory. The absorption pipettes are made in sets which are shaped in the form of globes, and a number of independent sets are required for the treatment of the different constituent gases. A simple pipette of the Hempel type is shown in Fig. 8.

The method of carrying on an analysis with the Hempel apparatus is as follows:

A sample of gas measuring 100 cc. is drawn into the burette, and then transferred to the first pipette, which contains potassium hydrate dissolved in two times its weight of water. This pipette absorbs carbon dioxide (CO_2). The gas is then passed into the second pipette, containing saturated bromine water, which absorbs the heavy hydrocarbons (C_2H_4); then into the third pipette, containing a solu-

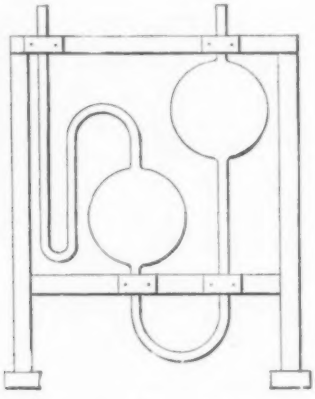


FIG. 8 HEMPEL PIPETTE

tion of pyrogallic acid and potassium hydrate in the proportion of 5 grams of acid to 100 cc. of hydrate, which absorbs oxygen (O); then into the fourth pipette, containing ammoniacal cuprous chloride, which absorbs carbonic oxide (CO), and finally into the fifth pipette, which is of large size and provided with exploding wires and galvanic battery, for the determination of marsh gas (CH_4) and hydrogen (H). A measured quantity of oxygen gas is added to this pipette and the contents exploded by an electric spark from the battery, resulting in a mixture of carbon dioxide, nitrogen, and free oxygen. The quantity of carbon dioxide is determined by passing the gas into the pipette containing potassium hydrate, and the quantity of oxygen by subsequently passing it into the pipette containing potassium pyrogallate, finally determining the quantity of marsh gas and hydrogen from the known reactions which occur during this process, and the composition of the resulting gases.

For each of these processes the pipettes are shaken to hasten the absorption, and the quantity absorbed is determined by returning the

gas into the measuring burette and observing the successive differences.

Tar and Soot.

The quantity of tar and soot contained in producer gas may be found by drawing a measured volume of gas through filter paper, weighing the paper before and after.

APPENDIX NO. 16

RINGELMANN CHART

A Ringelmann smoke chart is shown with full-size spacing in Fig. 9*. To use this chart, four cards are ruled like those shown, though covering a much larger area, and placed in a horizontal row about 50 ft. from the observer, and in line between him and the chimney, together with two other cards, one of which is white and the other solid black. The observer glances rapidly from the chimney to the cards and judges which one corresponds with the color and density of the smoke. He makes these observations every minute, or oftener if desired, recording the number of the card representing the character of the smoke at the instant of observation. The results are then plotted on a chart, and the variations shown graphically.

The lines in cards 1 to 4 are respectively 1, 2.3, 3.7, and 5.5 mm. thick, and the spaces 9, 7.7, 6.3, and 4.5 mm.

APPENDIX NO. 17

SMOKE METER

As an illustration of the soot-collecting method referred to on page 1699, which was carried out in a certain trial, a plate ⅞ in. wide and 24 in. long, representing a surface of 21 sq. in., was inserted through a hole in the top of the flue and suspended by a wire, the hole being covered after the plate was inserted. The plate was temporarily withdrawn every two hours during the progress of the test, and the collection of soot removed by a brush. This was tried on a boiler fitted with an automatic stoker and on another fitted with a smoke-burning furnace, and it was found that under various conditions as to character of coal and other changes, the weight of soot collected per hour varied from 9 to 184 mg.

APPENDIX NO. 18

USE AND CALIBRATION OF INDICATORS

The indicated horsepower should be determined from the average mean effective pressure of diagrams taken at intervals of 20 minutes, and at more frequent intervals if the nature of the test makes this necessary, this being done for each end of each cylinder. With variable loads, such as

* Reproduced from Trans. Am. Soc. M. E., vol. 21, p. 98, fig. 6.

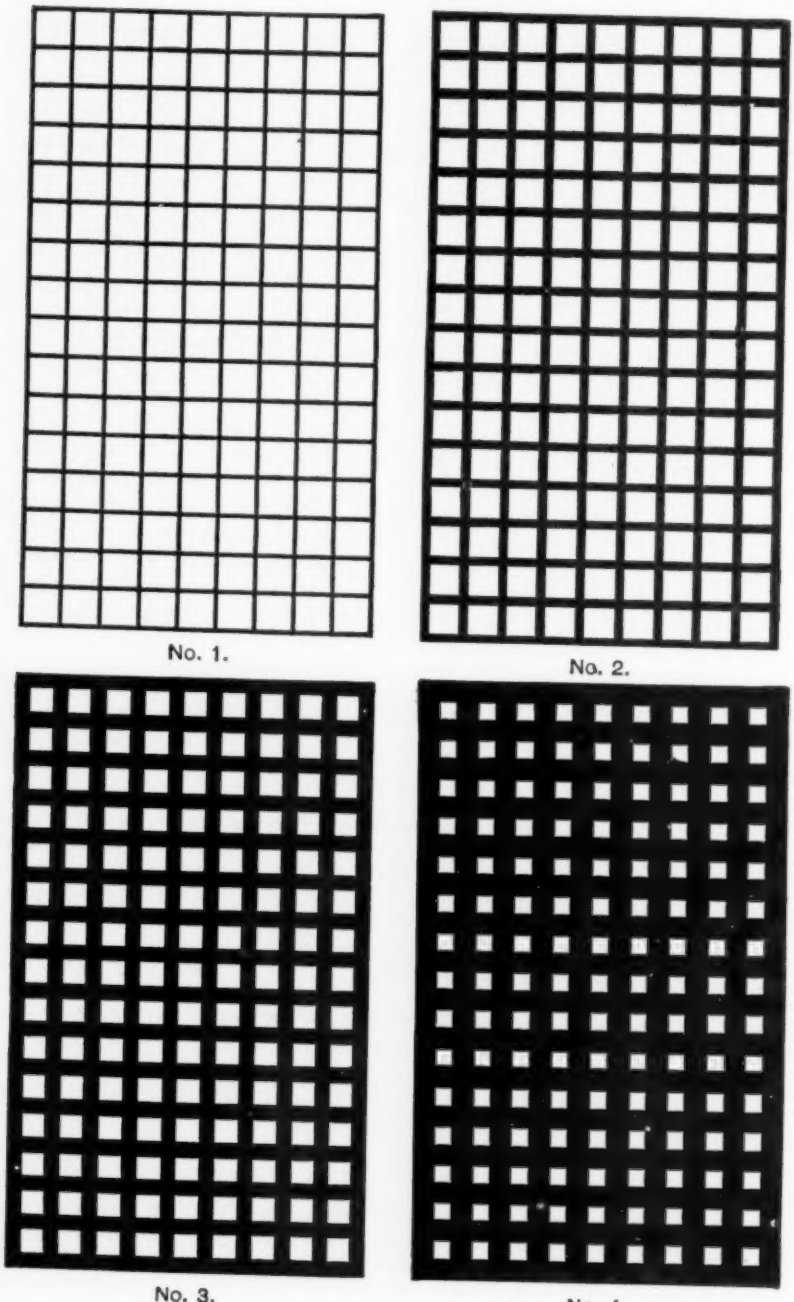


FIG. 9 RINGELMANN SMOKE CHART

those of engines driving generators for electric railroad work, and of rubber-grinding and rolling-mill engines, the diagrams cannot be taken too often. In cases like the latter, one method of obtaining suitable averages is to take a series of diagrams on the same blank card without unhooking the driving cord, and apply the pencil at successive intervals of 10 seconds until 2 minutes' time or more has elapsed, thereby obtaining a dozen or more indications in the time covered. This tends to insure the determination of a fair average for that period. In taking diagrams for variable loads, as indeed for any load, the pencil should be applied long enough to cover several successive revolutions, so that the variations produced by the action of the governor may be properly recorded. To determine whether the governor is subject to what is called "racing" or "hunting," a variation diagram should be obtained; that is, one in which the pencil is applied a sufficient time to cover a complete cycle of variations. When the governor is found to be working in this manner, the defect should be remedied before proceeding with the test.

It is seldom necessary, as far as average power measurements are concerned, to obtain diagrams at precisely the same instant at the two ends of the cylinder, or at the same instant on all the cylinders, when there are more than one. All that is required is to take the diagram at regular intervals. Should the diagrams vary so much among themselves that the average may not be a fair one, it signifies that they should be taken more frequently, and not that special care should be employed to obtain the diagrams of each set at precisely the same time. When diagrams are taken during the time when the engine is working up to speed at the start, or when a study of valve setting and steam distribution is being made, they should be taken at as nearly the same instant as practicable. In cases where the diagrams are to be taken simultaneously, the best plan is to have an operator stationed at each indicator. This is desirable, even where an electric or other device is employed to operate all the instruments at once; for unless there are enough operators, it is necessary to open the indicator cocks some time before taking the diagrams and run the risk of clogging the pistons and heating the high-pressure springs above the ordinary working temperature.

To determine the power developed while an engine is starting from rest and attaining working speed and working load, a number of diagrams should be taken in rapid succession, the speed counted for each one, and the results averaged. If the period of time thus covered on a commercial test is unduly long so as to be an appreciable percentage of the total running time, this power should be included in the average for the whole run, due regard being had for the time it is in operation. In that event the total duration of the run should be the whole time the throttle valve is open.

The most satisfactory driving rig for indicating is some form of well-made pantagraph, with driving cord of fine annealed wire leading to the indicator. The reducing motion, whatever it may be, and the connections to the indicator, should be so perfect as to produce diagrams of equal length when the same indicator is attached to either end of the cylinder, and produce a proportionate reduction of the motion of the piston at every point of the stroke, as proved by test.

With a perfect working pantagraph, or similar apparatus, the equality in the length of diagrams taken with the same indicator at the two ends is sufficient indication of the substantial reliability of the reduction when the point of cut-off on the diagram is not unusually short, say, not shorter than one-eighth. When the cut-off is unusually short, the error produced by imperfect reducing motion, stretch of cord, or otherwise, becomes a comparatively large item, and one which for accurate work should be allowed for. To test the accuracy

of the reducing motion without making special preparations for a thorough examination, it is sufficient to make a comparison between the actual proportion of the stroke covered and the apparent proportion measured on the indicator, and see how they agree. This may be done on a large engine by making the comparison wherever it happens to stop, and repeating the comparison when it has stopped with the piston at some other point of the stroke. With an engine which can be turned over by hand, or where auxiliary power is provided for moving it, the comparison may be made at a number of equi-distant points in the stroke. To make the test properly, a diagram should be taken just before stopping, and this will serve as a reference for the measurements taken after stopping. The actual proportion of stroke covered is determined by measuring the distance which the crosshead has moved and comparing it with the whole length of the stroke, making sure that the slack has all been taken up by turning sufficient steam into the cylinder to bring a pressure to bear on the piston, but not sufficient to start the flywheel in motion. To obtain the apparent indication from the diagram, the indicator pencil is moved up and down with the finger, so as to make a vertical mark on the diagram, and the distance of this mark from the beginning of the diagram compared to the whole length is the proportion desired.

It is necessary, of course, to go through these operations without changing in any way the adjustment of the driving cord of the indicator, or any part of the mechanism that would alter the movements of the indicator.

In all cases the pipes leading to indicators should be as short and direct as possible. The use of a three-way cock and a single indicator connected to the two ends of the cylinder is not advised, except in cases where it is impracticable to use an indicator close to each end. If a three-way cock is used, the error produced should be determined and allowed for.

The effect of the error produced by a three-way cock is usually to increase the area of the diagram. This is due to the tardiness of the indicator in responding to the changes of pressure. In an investigation which was carried out both on short-stroke engines running at high speed and long-stroke engines running at comparatively slow speed, it was found that the increased area of the diagram, due to the sluggish action referred to, ranged from 3 to 7 per cent as compared with an indicator having a short and direct pipe.

The point selected for attaching the indicator pipe to the cylinder should never be the drip pipe or any point where the water of condensation will run into the instrument, if this can possibly be avoided. The admission of water with the steam may greatly distort the diagram. If it becomes necessary to place the indicator in such a position, as may happen when it is attached to the lower end of a vertical cylinder, the connection to the indicator must be short and direct, and in some cases it should be provided with a drip chamber arranged so as to collect the water or deflect it from entering the instrument.

The pipe connections for indicating gas and oil engines should be removed as far as possible from the ports and ignition devices, and made preferably in the cylinder head. The pipes should be as short and direct as possible. Avoid the use of long pipes, otherwise explosions of the gas in these connections may occur.

Ordinary indicators suitable for indicating steam engines are much too lightly constructed for gas and oil engines. The pencil mechanism, especially the pencil arm, needs to be very strong to prevent injury by the sudden impact at the instant of explosion. A special gas-engine indicator is required for satisfactory work, with a small piston and a strong spring.

In the manipulation of the indicator it is important to keep the instrument in clean condition and preserve it in mechanically good order. Ordinary cylinder oil is the best material to use for lubricating the indicator piston for pressures above the atmosphere. It is better to have the piston fit the cylinder rather loosely, so as to get absolute freedom of motion.

than to have a mathematically accurate fit. In the latter case, extreme care and frequent cleanings are required to obtain good diagrams.

No diagrams should be accepted in which there is any appearance of want of freedom in the movement of the mechanism. A ragged or serrated line in the region of the expansion or compression lines is a sure indication that the piston or some part of the mechanism sticks; and when this state of things is revealed the indicator should not be trusted, but the cause should be ascertained and a suitable remedy applied. Entire absence of wire drawing of the steam line, and especially a sharp, square corner at the beginning of the steam line, should be looked upon with suspicion, however desirable and satisfactory these features might otherwise be. These are frequently produced by an indicator which is defective owing to want of freedom in the mechanism. An indicator which is free when subjected to a steady steam pressure, as it is under a test of the springs for calibration, should be able to produce the same horizontal line, or substantially the same, after pushing the pencil down with the finger, as that traced after pushing the pencil up and subsequently tapping it lightly. When the pencil is moved by the finger, first up and then down, the piston being subjected to pressure, the movement should appear smooth to the sense of feeling.

To make a comparison of indicator springs with standards, the calibration should be made, if this were practical, under the same conditions as those pertaining to their ordinary use. Owing to the fact that for steam work the pressure of the steam in the indicator cylinder and the corresponding temperature are undergoing continual changes, it becomes almost impossible to compare the springs with any standard under such conditions. There must be a constant pressure during the time that the comparison is being made. To bring the conditions for steam work as nearly as possible to those of the working indicator, the steam should be admitted to the indicator as short a time as practicable for each of the pressures tried, and then the indicator cock should be closed and the steam exhausted therefrom before another pressure is tried. By this means the parts are heated and cooled somewhat the same as under the working conditions. For each required pressure the first step is to open and close the indicator cock a number of times in quick succession, then quickly draw the line for the desired record, observing the gage or other standard at the same instant. A corresponding atmospheric line is taken immediately afterwards.

The calibration should be made for at least five equidistant points. For ordinary work the arithmetical mean of the various results should be taken for the average scale.

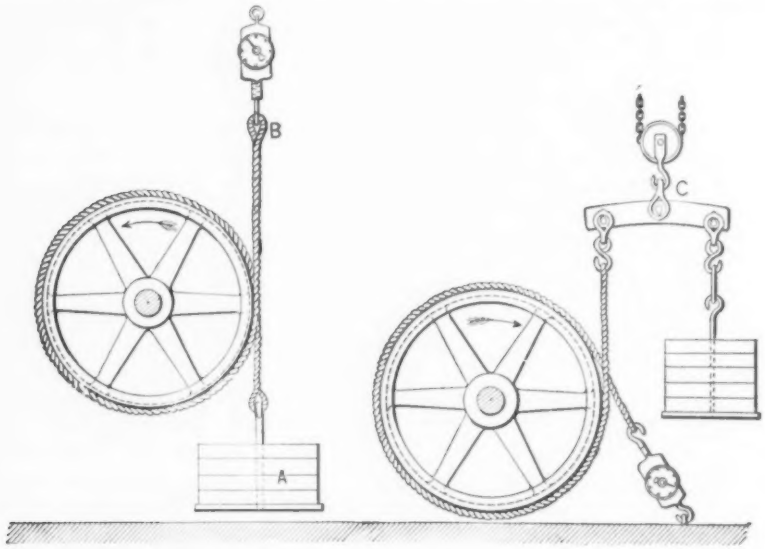
The indicator springs used for gas and oil engines should also be calibrated with the indicator in as nearly as possible the same condition as to temperature as exists during the trial. A simple way of heating the indicator is to subject it to a steam pressure just before calibration. Compressed air is suitable for the actual work of calibration, being used in preference to steam so as to bring the conditions as near as possible to those which obtain when the indicators are in actual use in such engines.

The standard of comparison in all cases is a steam gage which has been proved correct by reference to either a mercury column or a dead weight testing apparatus.

APPENDIX NO. 19

FRICTION BRAKES

A self-adjusting rope brake is illustrated in Fig. 10* in which it will be seen that if the friction at the rim of the wheel increases, it will lift the weight *A*, which action will diminish the tension in the end *B* of the rope, and thus prevent a further increase in the friction. The same device can be used for a band brake of the ordinary construction. Where space below the wheel is limited, a cross bar, *C*, supported by a chain tackle exactly at its central point, may be used as shown in Fig. 11*, thereby causing the action of the weight on the brake to be upward. A safety stop should be



FIGS. 10 AND 11 ROPE BRAKES

used with either form, to prevent the weights being accidentally raised more than a certain amount.

A water-friction brake is shown in Fig. 12*. It consists of two circular disks, *A* and *B*, attached to the shaft *C*, and revolving in a case *E*, between fixed planes. The space between the disks and planes is supplied with running water, which enters at *D* and escapes at the cocks *F*, *G*, and *H*. The friction of the water against the surfaces constitutes a resistance which absorbs the desired power, and the heat generated within is carried away by the water itself. The water is thrown outward by centrifugal action and fills the outer portion of the case. The greater the depth of the

* Reproduced from Trans. Am. Soc. M. E., vol. 24, pp. 739 and 740, figs. 119. 120, 121.

ring of water, the greater the amount of power absorbed. By adjusting suitably the amount of water entering and leaving any desired power can be obtained.

Another form of water brake is that shown in Fig. 13, which is used by the Westinghouse Machine Company for large turbines. In this brake the shaft is carried in two bearings and is coupled to the turbine shaft. The revolving paddle wheel or "runner" is designed to run in very close clearance with the serrated-rim piece. The outer casing of the brake maintains

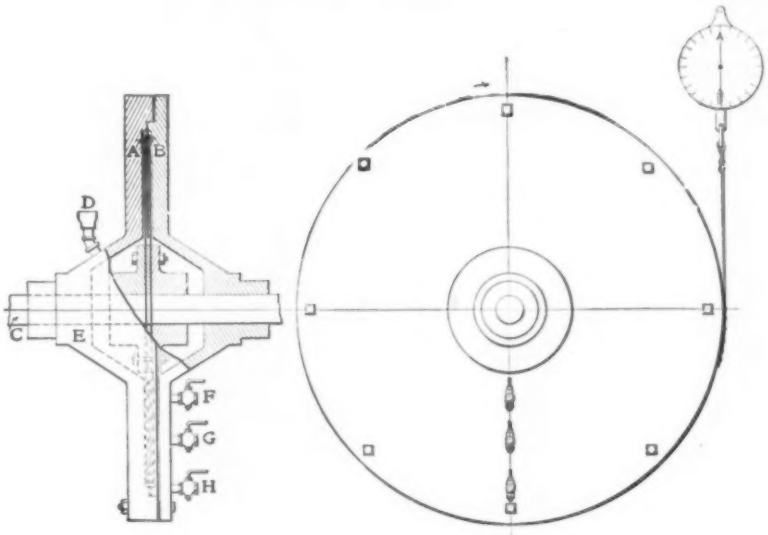


FIG. 12 WATER BRAKE

its accurate position surrounding the runner by means of carefully constructed bearings. To this casing is rigidly attached the lever arm, made of such length as to facilitate ready calculation. The little roller wheel on the end of the arm bears upon platform scales which weigh the load.

Water is introduced through a hose connection at the opening marked *B*. The water enters the interior of the runner, is thrown out by centrifugal force through small holes drilled in the rim into the outer teeth spaces where a resistance is created, and it escapes with difficulty through the close clearance around the outside of the runner, finally discharging at the bottom through openings *C* (Fig. 12) and *C* (Fig. 13), where it runs to waste.

For a description of the Alden water brake, see the paper on An Automatic Absorption Dynamometer.[1]

The application of water brakes for making field tests of waterwheels is shown in Professor Allen's paper on The Testing of Water Wheels After Installation.[2]

[1] Geo. I. Alden. Trans. Am. Soc. M. E., vol. 11, p. 958.

[2] C. M. Allen. Trans. Am. Soc. M. E., vol. 32, p. 275.

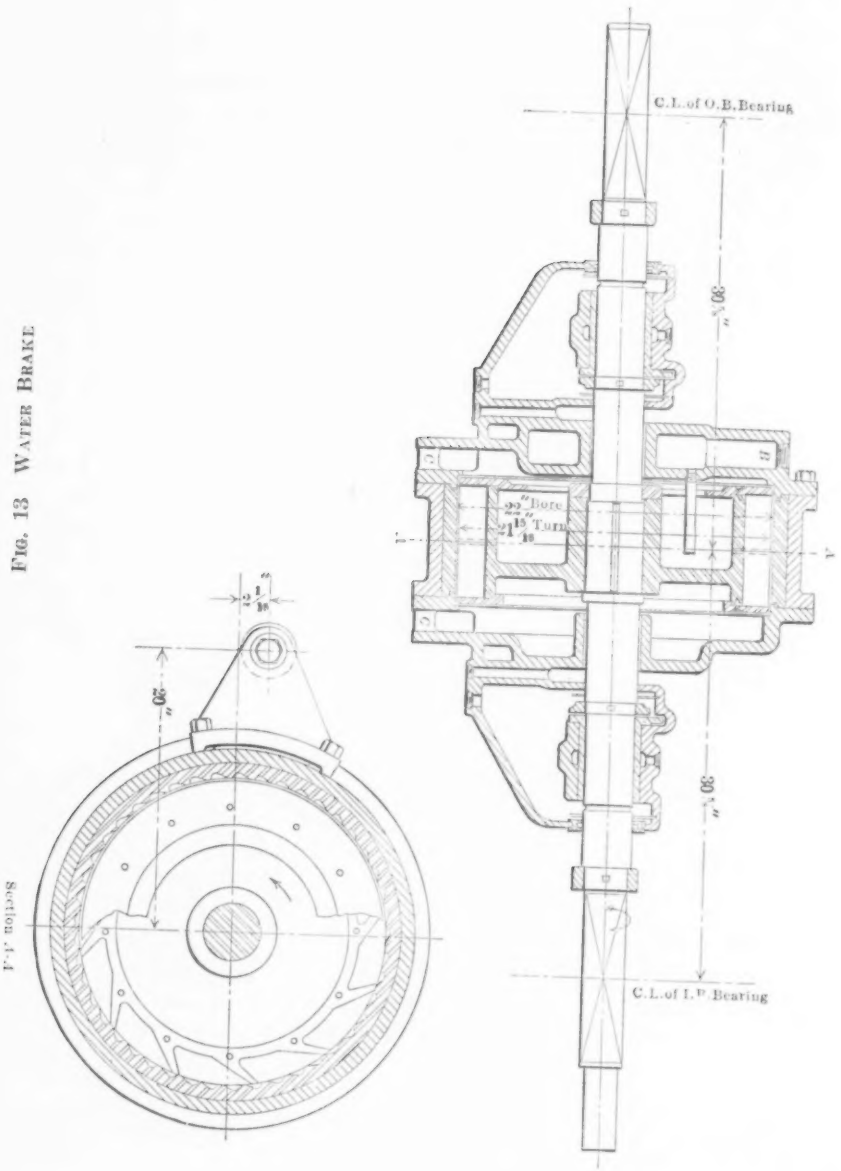


FIG. 13 WATER BRAKE

APPENDIX NO. 20

DYNAMOMETERS

A form of dynamometer for measuring the power transmitted from an engine to a mill shaft is well illustrated in principle by the Webber balance dynamometer, shown in Vol. 4 of the Transactions.[1] In this apparatus, which consists essentially of a compound gear, the intermediate gear is carried by a weighted arm, which is free to rise and fall, according to the amount of force transmitted. The power is determined from the net weight thus lifted.

Another form which is useful for mill purposes is the belt dynamometer, the principle of which is shown in Fig. 14. *A* is the driving pulley and *B* the driven pulley. The connecting belt passes over the dynamometer pulleys *C* and *D*. These two pulleys are mounted on a frame carrying a scale beam *F*, all of which turns on the center *E*. The difference in the total strain on the two sides of the belt is computed by multiplying the net weight on the beam by the leverage, the latter being found by dividing the

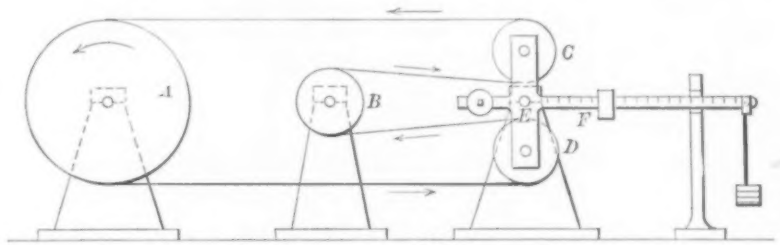


FIG. 14 BELT DYNAMOMETER

length of the beam by the distance from the center *E* to the centers of pulleys *C* and *D*.

The *shaft dynamometer* consists of the following essential parts: a long tube encircling the shaft and made fast thereto at one end, being free at the other end and maintained in alignment by adjustable rollers; two radial arms, one attached to the shaft and the other to the free end of the tube, which rotate a slight amount with reference to each other, according to the twist of the enclosed length of shaft; and a set of levers which multiply this rotative movement and at the same time convert it into linear motion, which is transmitted to a sleeve and collar mounted upon the shaft and sliding thereon. These parts all revolve with the shaft. An independent indicating apparatus is provided, which is mounted on a stationary frame, and the sliding movement of the rotating collar is transmitted through it to an index hand. The torsional strain is determined from the reading of the accompanying scale, which is graduated to millimeters.

The zero reading is found by disconnecting the propeller and turning the shaft at a slow speed, first in one direction and then in the other, observing the indication in both cases, and fixing the point of zero strain at the mean of the two. When it is impracticable to disconnect the propeller the read-

[1] S. S. and W. O. Webber. Trans. Am. Soc. M. E., vol. 4, p. 227.

ings may be taken when the vessel is drifting under her own headway after shutting off steam. The calibration of the instrument, which can best be done when the shaft is in the shop before installation, is carried on by securing the shaft in a fixed position, and applying a torsional strain by means of weights at the end of a lever attached beyond the dynamometer, taking readings with a number of different weights.

The horsepower shown by the dynamometer is determined by multiplying the reading of the instrument expressed in millimeters by the number of revolutions of the shaft per minute, and by a constant determined from the calibration noted, the constant being an expression for the horsepower corresponding to 1 r.p.m. and a reading of 1 mm.

A shaft dynamometer requires delicate manipulation, a load of 500 h.p. in some instruments causing a movement at the end of the two arms of only 1/100 of an inch.

The *locomotive dynamometer*, as formerly designed, consists essentially of one or more strong helical springs placed between the drawbar of the locomotive and the train, and the distance which the spring is extended furnishes a measurement on an appropriate scale of the amount of force transmitted, and thereby the amount of power. The strain on the spring is transmitted through mechanism to a registering apparatus placed in the dynamometer car, and a continuous record is made upon a strip of paper traveling at a definite rate of speed. The apparatus used by the Pennsylvania R. R., the Chicago, Burlington and Quincy R. R., and the Chicago, Milwaukee and St. Paul R. R., are shown in the cuts given on pages 1329 to 1334 of Transactions, Vol. 14.

The latest form of dynamometer car makes use of oil plungers in place of springs, and employs oil transmission for the recording devices.

APPENDIX NO. 21

USE AND CALIBRATION OF ELECTRICAL INSTRUMENTS

To determine whether the readings of any instrument are disturbed by stray fields, the instrument is turned bodily through an angle of 180 deg. If there are stray fields, the change of position will change the reading; in which case the instrument should be moved to some other point where the reading is found to be unaffected.

The proper methods of connecting wattmeters in polyphase alternating-current systems are shown in Figs. 15 and 16. The former refers to two-phase and the latter to three-phase systems. In Fig. 15 there are four cables, designated *A*, *B*, *C*, and *D*, arranged in two pairs, or one pair for each phase. Each wattmeter is connected to its respective pair of cables, the current coil being in series with one of the two cables, and the potential coil in parallel with the load or bridging the two cables. In Fig. 16 there are three cables, which for convenience are designated by the letters *X*, *Y*, and *Z*. Meter W_1 is connected to the phase represented by the cables *X* and *Y*, and meter W_2 to that represented by the cables *Y* and *Z*. The current coil of the former is connected in series with cable *X*, the potential coil bridging *X* and *Y*. The current coil of the latter is connected in series with cable *Z*, the potential coil bridging *Y* and *Z*. The connections

shown are elementary; transformers and other equipment usually employed in actual installations being omitted. The total output is the arithmetical sum of the quantities indicated by the two meters for a two-phase system, and the algebraic sum for a three-phase system.

The power factor is the proportion borne by the true power or kilowatt output as shown by wattmeters, to the apparent power or kilovolt-ampere (kv. a.) output determined from the readings of the voltmeters and ammeters. In a single-phase system the kv.a. output is found by multiplying the volts by the amperes and dividing by 1000. In a two-phase system it is found by multiplying the volts by the amperes in each phase, adding the two products together, and dividing by 1000. In a three-phase system it is ascertained by multiplying the average volts of the three circuits by

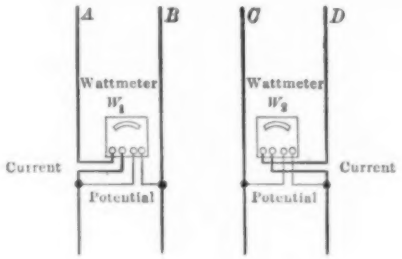


FIG. 15 TWO-PHASE SYSTEM

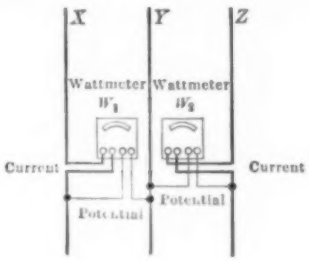


FIG. 16 THREE-PHASE SYSTEM

the average amperes of the three circuits, multiplying the product by the constant 1.73, and dividing by 1000.

In a three-phase system which is balanced (that is, when there is an equality of amperes and volts in all three phases), the power factor may also be determined by referring to the curve shown in Fig. 17, in which abscissae represent the proportion borne by the readings of the two wattmeters, and ordinates the corresponding power factor.

When the power factors in a three-phase system is 100 per cent, both wattmeters indicate the same quantity. When it is less than 100 per cent, one of the meters indicates a larger quantity than the other, the one showing the larger quantity giving positive readings, and the other either positive or negative readings. The latter become negative when the power factor falls below 50 per cent. To determine whether the smaller reading is to be considered negative or not, disconnect the potential circuit of the meter showing the smaller reading from the cable *Y* (referring to Fig. 16), and connect it to the opposite outside cable so that this circuit will bridge the two outside cables. If the meter reverses, i.e., if the index hand falls back to the zero point and rests against the stop, the readings taken with the meter in its working position must be considered negative. If it does not reverse, the readings are both positive.

A watt-hour meter can best be calibrated by reference to an indicating wattmeter, in accordance with the following directions:

Ascertain first the constants and formula pertaining to the watthour meter by referring to the dial of the instrument, and, if need be, to the manufacturer's instruction book; and by this means determine the number of watts corresponding to one revolution of the meter disk per second. Then find the actual speed of the disk by observing the time required for a certain number of complete revolutions, using a stop watch, and compute the actual number of revolutions per second; finally, multiplying the revolutions per second by the watts corresponding to one revolution per second, and the result is the output in watts. Read the indicating wattmeter every five seconds during the period that the speed is observed, and average the readings. A comparison of the watts determined by the watt-hour meter observations with this average shows the error of the watt-hour meter.

Calibrations should be made while the main test is going on. If the load is

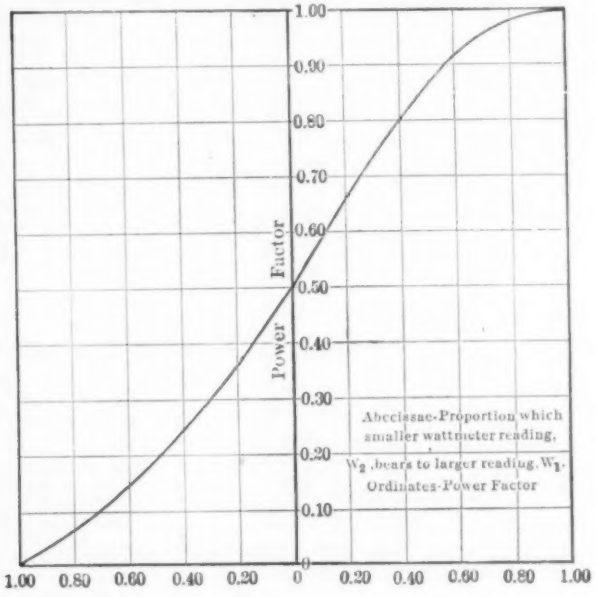


FIG. 17 POWER FACTOR CURVE

not substantially constant they should be repeated at regular intervals during its progress so as to obtain the comparison under average conditions.

Reference may be made to the Standardization Rules of the American Institute of Electrical Engineers for methods of determining generator temperatures and for general information bearing on electrical standards.

APPENDIX NO. 22

WATER RHEOSTATS

A simple form of water-rheostat, suitable for absorbing the output of a 500-kw. generator of the three-phase alternating-current type, consists of three wine casks, one for each phase, placed side by side, through

APPENDIX NO. 23

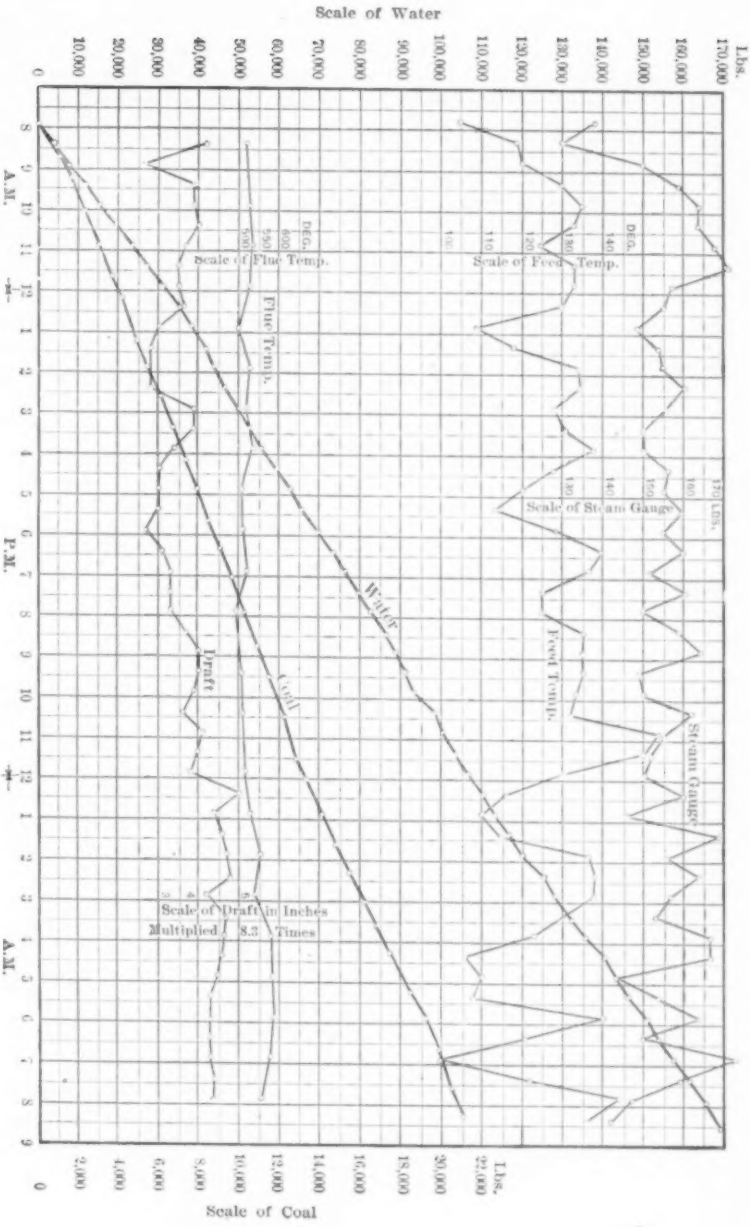


FIG. 18. CHART SHOWING LOG OF BOILER TEST

Reproduced from Trans. Am. Soc. M. E. vol. 21, p. 104, fig 7.

each of which a sufficient amount of cooling water is allowed to flow to prevent boiling; salt being added to the water as may be needed to lower its resistance. A fixed terminal is provided consisting of three connected iron plates, one of which is laid on the bottom of each cask. A movable terminal is provided for each phase, consisting of a piece of 2-in. standard iron pipe, 4 to 6 ft. in length, suspended vertically in the water at the center of each cask. Each pipe is arranged to be raised or lowered at will, so as to regulate the depth to which it is immersed, and thereby the output for that phase. By varying the relative immersion of the pipes, an equal distribution of the current may be obtained between the three phases.

Another arrangement consists of a tank, the size of which equals that of the three casks combined, the pipes being located a suitable distance apart and a single plate being used for the fixed terminal.

When a larger amount of current than 500 kw. is generated, the size of the tank and terminals may be proportionately increased, and at the same time provision made for introducing the cooling water so as to flow along the surface of each pipe and overcome or reduce the tendency to local ebullition.

The changes of resistance due to variations in the strength of the salt solution, depth of immersion of the terminals, and temperature of the water, are such that constant attention is required to maintain uniformity in the output with this kind of rheostat. When uniformity is desired without such attention a rheostat may be constructed which embodies the principles referred to in the description of the apparatus used for a direct current rheostat, which is given on Page 1701.

APPENDIX NO. 24

LOCATION OF INSTRUMENTS FOR BOILER TESTS

(*a*) The feedwater thermometer should be placed in a thermometer well and inserted in the feed pipe. Except in cases where an injector is used, the point selected should be as near as practicable to the boiler. Where an injector is employed, and the water is weighed or measured before it is supplied thereto, the well should be placed on the suction side of the injector, and the injector should receive steam through a short covered pipe connected directly to the boiler under test. If the steam is taken from some other source and it is of different pressure and different quality from that of the boiler under test, correction should be made for such difference, and especially for any excessive moisture thus introduced into the feedwater. When the temperature of the water changes between the injector and boiler, as by the use of a heater or by excessive radiation, the temperature at which the water not only enters and leaves the injector, but that also at which it enters the boiler, should all be taken. In that case, the weight to be used is that of the water leaving the injector, computed from the heat units if not directly measured and the temperature, that of the water entering the boiler. The weight of condensed steam to be added to the weight of water entering the injector, to obtain that leaving the injector, may be computed by multiplying the weight entering by the proportion

$$\frac{h_2 - h_1}{h_3 - h_3}$$

in which

h_1 = heat units per lb. of water entering injector
h_2 = heat units per lb. of steam entering injector
h_3 = heat units per lb. of water leaving injector

(*b*) The location of the steam calorimeter and steam thermometer should be as close to the boiler as possible, keeping in mind the directions given on page 1713.

(*c*) Draft gages should be attached to each boiler between the hand damper and the boiler, and as near the damper as practicable. In the case of a plant containing a number of boilers, a gage should also be attached to the main flue between the regulating damper and the boiler plant. It is desirable also to have gages connected to the furnace or furnaces of the boilers, and in cases of forced blast, to the ashpits and blower ducts. If there is an economizer in the flue a gage should be connected to the flue at each end of this apparatus. The same draft gage may be used for all the points noted, provided suitable pipes are run from the gage to each, arranged so as to be readily connected to either point at will.

(*d*) The flue thermometer should be located where it will show the average temperature of the whole body of gas. For an extremely large flue the thermometer may be placed in an oil pot of small diameter, which is suspended in the flue, and the thermometer lifted partially out of the oil when the temperature is read.

(*e*) The sample for flue gas analysis should be drawn from the region near the center of the main body of escaping gases, and the point selected

should be one where there is no chance for air leakage into the flue, which could affect the average quality. In a round or square flue having an area of not more than one-eighth of the grate surface, the sampling pipe may be introduced horizontally at a central point, or preferably a little higher than the central point, and the pipe should contain perforations extending the whole length of the part immersed and pointing toward the current of gas, the collective area of the perforations being less than the area of the pipe.

APPENDIX NO. 25

DETERMINATION OF HEAT CONSUMPTION OF ENGINES AND TURBINES

The measurement of the heat consumption requires the measurement of each supply of feedwater to the boiler—that is, the water supplied by the main feed pump, that supplied by auxiliary pumps, such as jacket water, water from separators, drips, etc., and water supplied by gravity or other means; also the determination of the temperature of the water supplied from each source, together with the pressure and quality of the steam.

The temperatures at the various points should be those applying to the working conditions. It frequently happens that the measurement of the water requires a change in the usual temperature of supply. For example, where the main supply is ordinarily drawn from a hot-well in which the temperature is, say 100 deg. fahr., it may be necessary, owing to the low level of the well, to take the supply from some source under a pressure or head sufficient to fill the weighing tanks used, and this supply may have a temperature much below that of the hot-well; possibly as low as 40 deg. fahr. The temperature to be used is not the temperature of the water as weighed in this case, but that of the working temperature of the hot-well. The working temperature in cases like this must be determined by a special test, and included in the log sheets.

In determining the working temperatures, the preliminary or subsequent test should be continued a sufficient time to obtain uniform indications, and such as may be judged to be an average for the working conditions. In this test it is necessary to have some guide as to the quantity of work being done, and for this reason the power developed by the engine should be determined by obtaining a full set of diagrams at suitable intervals during the progress of the trial, and in a turbo-generator, the output of the generator should be observed. Observations should also be made of the gages connected with the plant and of the water levels in the boilers, the latter being maintained at a uniform point so as to be sure that the rate of feeding during the test is not sensibly different from that of the main test.

When the feedwater is all supplied by one feeding instrument, the temperature to be found is that of the water in the feed pipe near the point where it enters the boiler. If the water is fed by an injector this temperature is to be corrected for the heat added to the water by the injector, and for this purpose the temperatures of the water entering and of that leaving the injector are both observed. If the water does not pass through a heater on its way to the boiler (that is, that form of heater which depends upon the rejecting heat

of the engine or turbine, such as that contained in the exhaust steam either of the main cylinders or turbine or of the auxiliary pumps) it is sufficient, for practical purposes, to take the temperature of the water at the source of supply, whether the feeding instrument is a pump or an injector.

When there are two independent sources of feedwater supply, one the main supply from the hot-well, or from some other source, and the other an auxiliary supply derived say from the water condensed in the jackets of the main engine and in the live-steam reheater, if one be used, they are to be treated independently. The remarks already made apply to the first, or main, supply. The temperature of the auxiliary supply, if carried by an independent pipe either direct to the boiler or to the main feed pipe near the boiler, is to be taken at a convenient point in the independent pipe.

When a separator is used in the main steam pipe, arranged so as to discharge the entrained water back into the boiler by gravity, no account need be made of the temperature of the water thus returned. Should it discharge either into the atmosphere to waste, to the hot-well, or to the jacket tank, its temperature is to be determined at the point where the water leaves the separator before its pressure is reduced.

When a separator is used, and it drains by gravity into the jacket tank, this tank being subjected to boiler pressure, the temperatures of the separator water and jacket water are each to be taken before their entrance to the tank.

The heat to be determined is that used by the entire engine or turbine equipment, embracing the main cylinders or turbine and all auxiliary cylinders and mechanism concerned in the operation of the engine, including the air pump, circulating pump, and feed pumps, also the jacket and reheater when these are used. No deduction is to be made for steam used by auxiliaries unless these are shown by test to be unduly wasteful. In this matter an exception should be made in cases of guarantee tests where the engine contractor furnishes all the auxiliaries referred to. He should, in that case, be responsible for the whole, and no allowance should be made for inferior economy, if such exists. Should a deduction be made on account of the auxiliaries being unduly wasteful, the method of waste and its extent, as compared with a standard of known value, should be definitely noted.

The steam pressure and the quality of the steam are to be taken at some point conveniently near the throttle valve. The quantity of steam used by the calorimeter must be determined and properly allowed for.

APPENDIX NO. 26

BRITISH STANDARD OF ENGINE ECONOMY

The British standard, which gives the heat consumption per minute per unit of power, relates to the heat calculated from an assumed temperature of the feedwater, which is taken as that corresponding to the back pressure steam, allowance being made for heat derived from the drips of jackets, reheaters, etc.

As to the adaptability of the British standard, as compared with the standard given in this code, it should be borne in mind that the practical utility of an engine depends almost wholly upon the fact that it is used for some form of industrial work, and, in combination with a boiler and various appurtenances, it is a part of a complete power plant. Were it not for the unreliable character of coal and other fuels, the proper standard of economy for such an engine,

with a boiler of given efficiency, would be the fuel consumed, because fuel is, in reality, the source of the power, and this is the most important thing to be supplied in operating the engine. The use of a standard of heat units in place of fuel, meets the objectionable characteristics of coal due to its heat variability, and in doing this it makes no change in the conditions under which the standard should be employed. These are the actual conditions of use, and not the ideal conditions. The British method furnishes a means of determining and stating the ideal performance of an engine working under assumed conditions. It does not give the performance of the entire engine plant under the

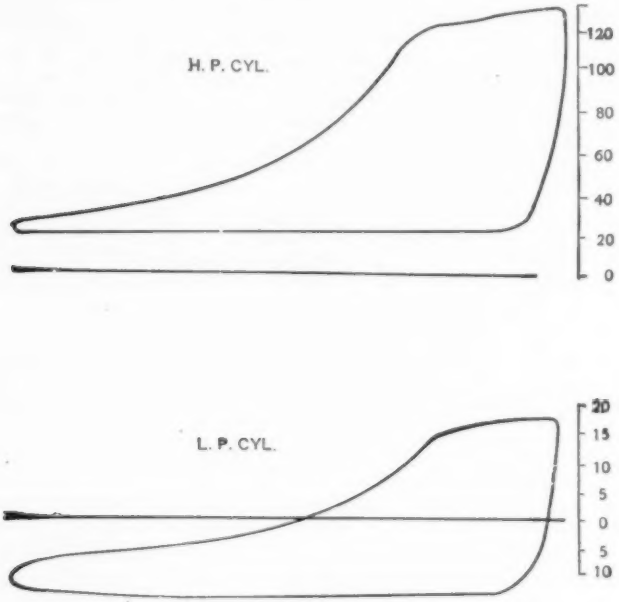


FIG. 19 ACTUAL DIAGRAM RECEIVER ENGINE

actual conditions as affected by the efficiency of the heaters and auxiliaries, although the performance under such conditions is usually of the greatest importance.

The British standard is useful in the following particulars:

(1) It expresses the economy of the engine as an independent machine, unaffected by the failings of the feedwater heater or condenser.

(2) In this form, it is useful for expressing the economy of an engine in the testing shop of an engine builder, or in the mechanical laboratory of a college, where it forms no part of a power plant.

(3) A comparison of the British standard with the economy of an ideal engine working between the same limits of temperature reveals those losses going on which are due to the engine alone, and this furnishes information not otherwise obtained.

NOTE: A method of studying cylinder performance by means of logarithmic diagrams has been proposed by J. Paul Clayton. See The Journal, Am. Soc. M. E., April 1912.

APPENDIX NO. 27

THE COMBINED DIAGRAM

The Combined Diagram is a hypothetical figure, which in its essential features represents an indicator diagram which would be obtained if the whole process of admission, expansion and exhaust occurred in one cylinder, viz., the low-pressure cylinder. It is a diagram from which the pressure of the steam at any point in the stroke of either cylinder, and the volume of that steam can be measured from one diagram, in the same manner that it can be measured in the case of a single cylinder engine from the actual indicator diagram.

The general method of laying out a combined diagram is shown clearly in Figs. 19 to 22*, the first of which refers to a Corliss compound engine (receiver type) in which the cylinder ratio is 3.72, and the clearance 4 per cent and 8 per cent, respectively, and the second to a Westinghouse compound engine (Woolf type), in which the cylinder ratio is 2.72, and the clearances, 33 per cent and 9 per cent, respectively.

APPENDIX NO. 28

DIAGRAM FACTOR

The diagram factor is the proportion borne by the mean effective pressure measured from the actual diagram to that of an ideal diagram which represents the maximum power obtainable from the steam accounted for by the actual diagram at the point of cut-off; assuming first that the engine has no clearance; second, that there are no losses through wire-drawing the steam either during admission or release; third, that the expansion line is a hyperbolic curve; and, fourth, that the initial pressure is that of the boiler; and the back pressure that of the atmosphere for a non-condensing engine, and of the condenser for a condensing engine.

The method of determining the diagram factor is illustrated in Figs. 23 to 26*, which apply to a simple non-condensing engine, a simple condensing engine, and a compound condensing engine. In Fig. 23, *RS* represents the volume of steam at boiler pressure admitted to the cylinder, *PR* and *OS* being hyperbolic curves drawn through the compression and cut-off points respectively. In Fig. 24 the factor is the proportion borne by the area of the actual diagram to that of the diagram *CNHSK*. In Fig. 25 the factor is the proportion borne to the arc of the diagram *CNHSK*. In Fig. 26 the factor is the proportion borne by the area of the two combined diagrams to the area *CNHSK*. In Fig. 24 where the diagram is the same as in Fig. 23, the distance *CN* is laid off equal to *RS* shown in Fig. 23, and the curve *NH* is a hyperbola referred to the zero lines *CM* and *MJ*. In Fig. 25 the distance *CN* is found in a similar way. In Fig. 26 the distance *CN* for the high-pressure cylinder is found in the same manner as in the case of a simple engine. The mean effective pressure of the ideal diagram can readily be obtained from the formula

$$\frac{P}{R}\ (1 + \text{hyperbolic logarithm } R) - p$$

where

P = absolute pressure of steam in boiler

$R = \dfrac{MJ}{CN}$

p = pressure of atmosphere or that in condenser

* Reproduced from Trans. Am. Soc. M. E., vol. 24, pp. 746, 748, 753, 754, figs. 123-126, 130-133.

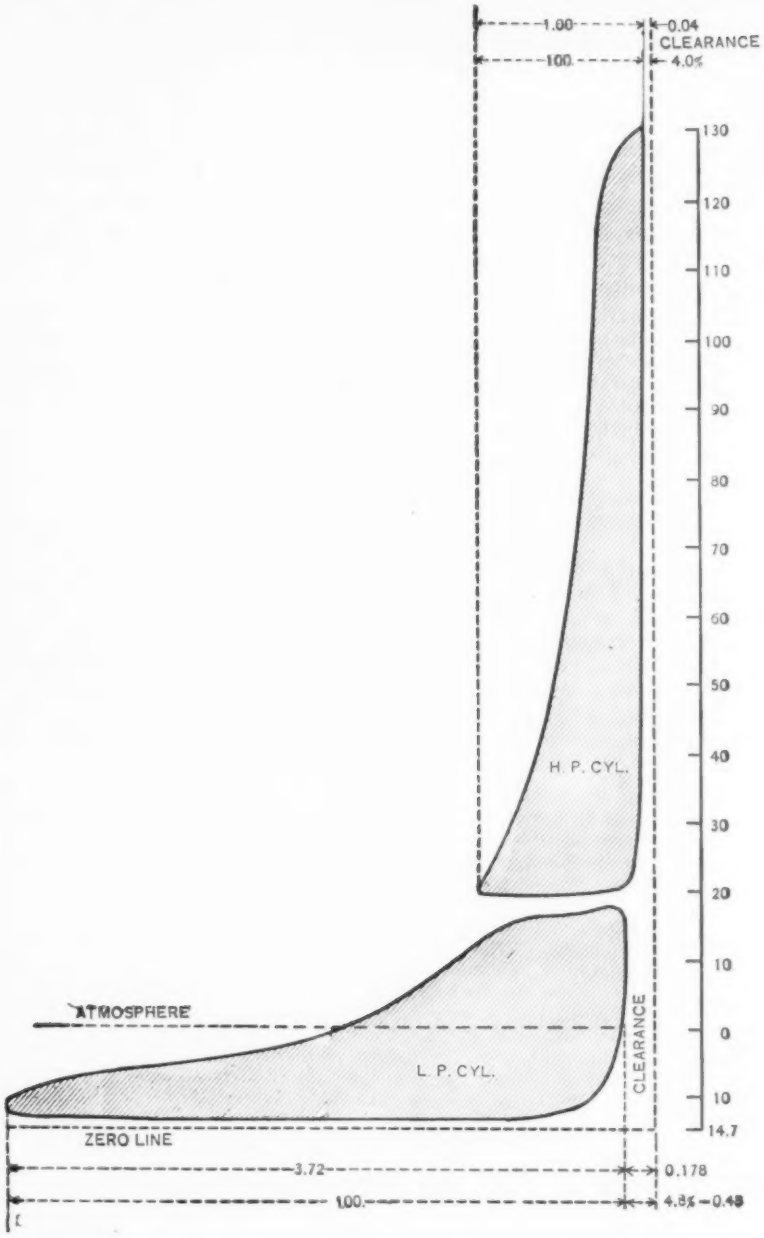


FIG. 20 COMBINED DIAGRAM RECEIVER ENGINE

FIG. 21 ACTUAL DIAGRAM WOOLF ENGINE

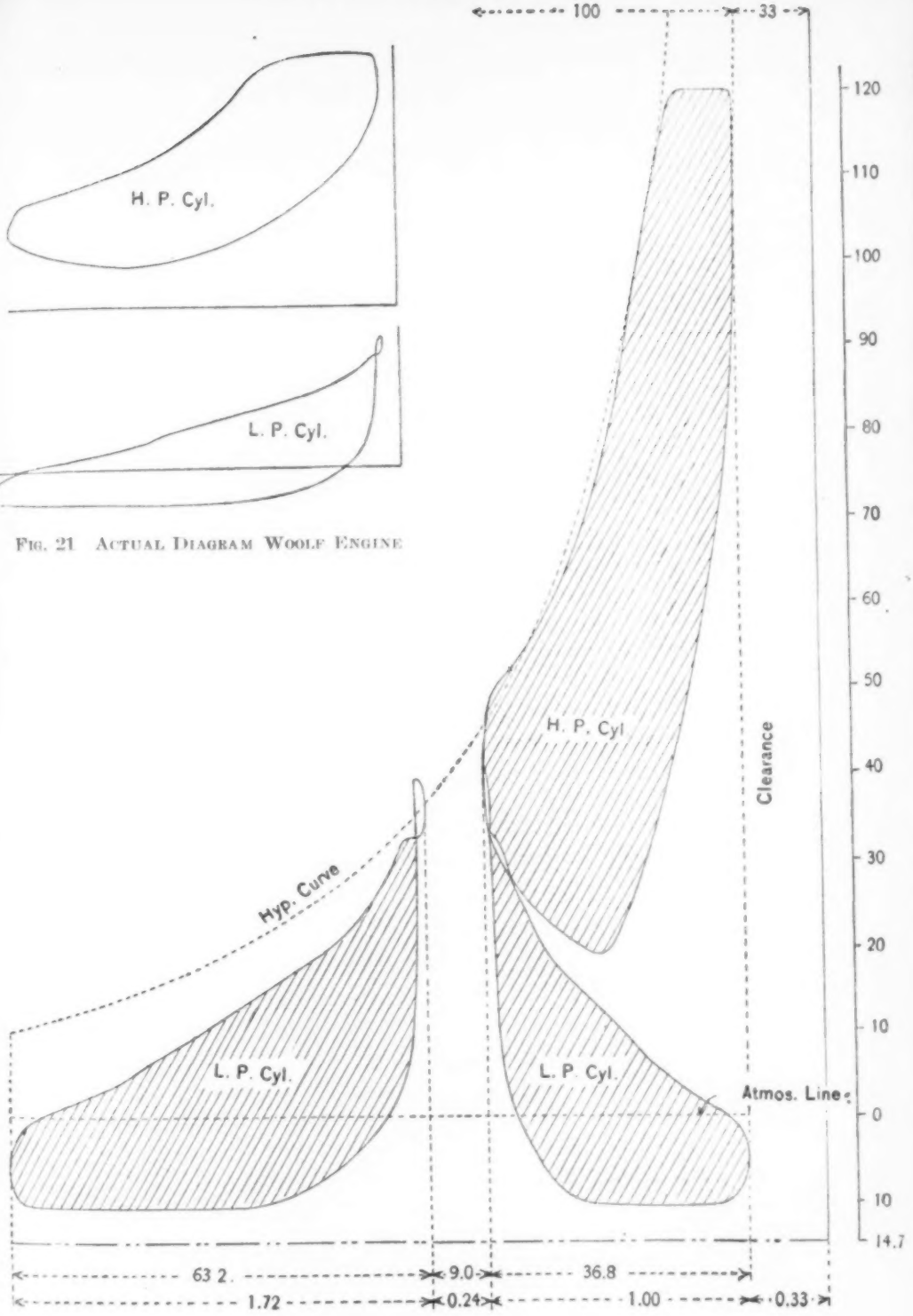


FIG. 22 COMBINED DIAGRAM WOOLF ENGINE

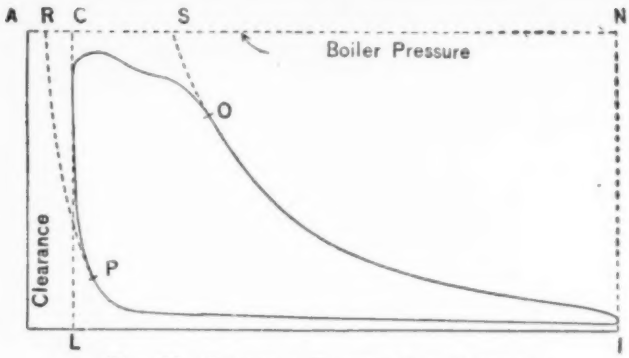


FIG. 23 DIAGRAM FACTOR NET VOLUME

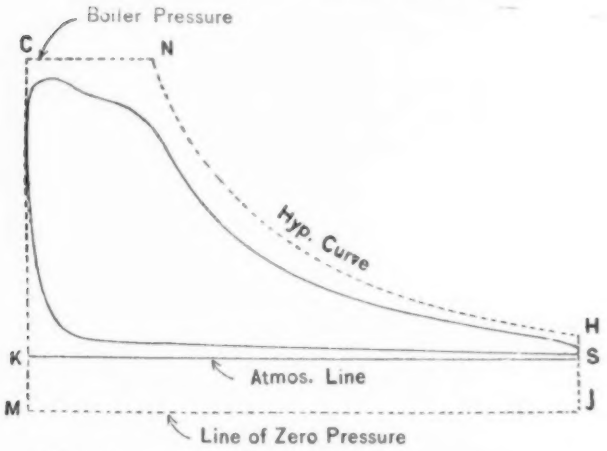


FIG. 24 DIAGRAM FACTOR NON-CONDENSING ENGINE

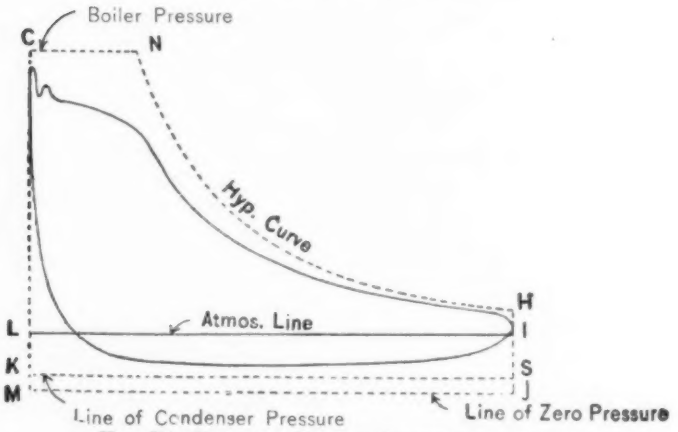


FIG. 25 DIAGRAM FACTOR CONDENSING ENGINE

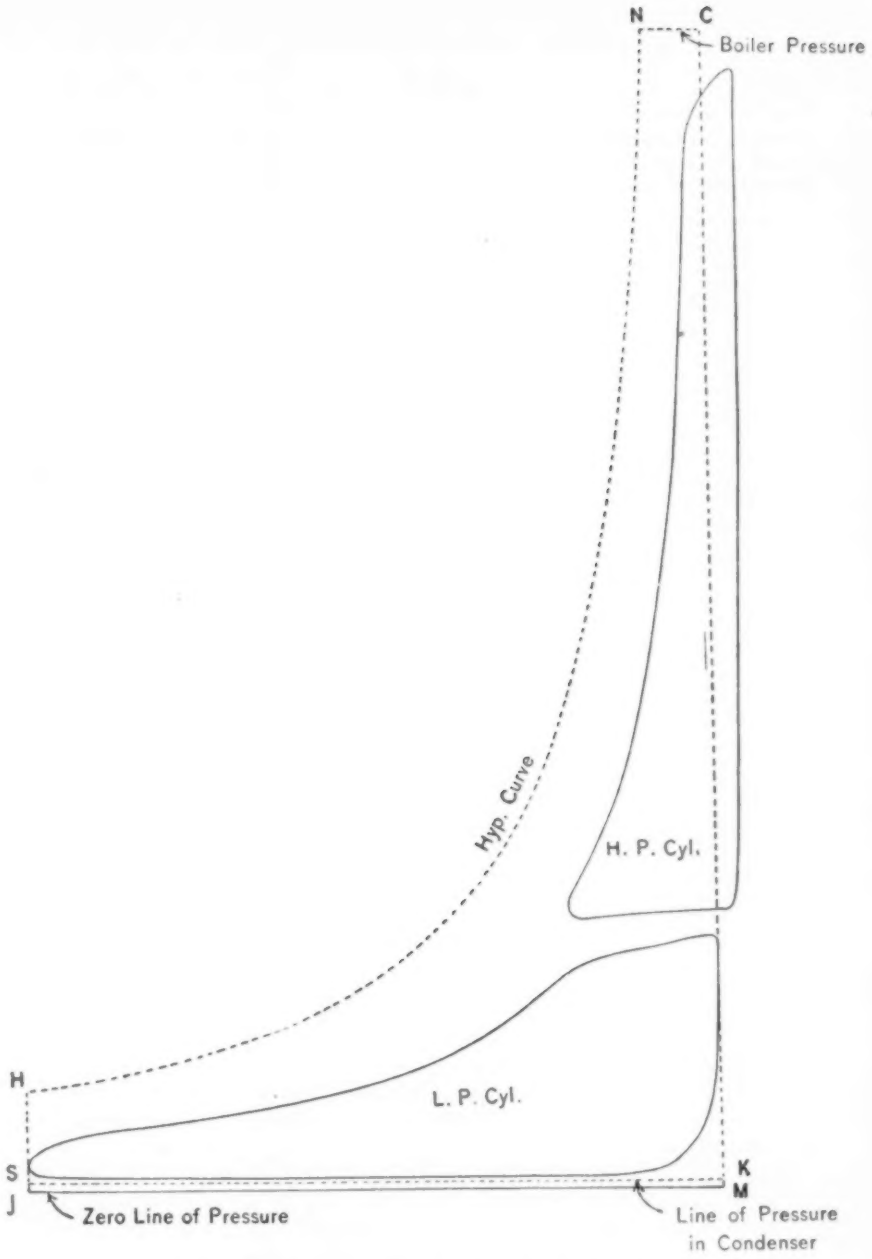


FIG. 26 DIAGRAM FACTOR COMPOUND ENGINE

APPENDIX NO. 29

HEAT ANALYSIS

Reference may be made to standard textbooks on thermodynamics for detailed information regarding the methods to be pursued and the formulæ to be used in working out the heat analysis in a steam engine required in research work. The principles involved may be briefly summarized as follows:

A certain amount of heat is supplied to the engine in a given time, and this is represented by the number of lb. of steam supplied, multiplied by its total heat. A part of this heat is used in the jacket, if one be employed, and the balance passes through the cylinder. That entering the jacket is lost partly by radiation from the outside surface, and the remaining portion enters the walls of the cylinder and is taken up by the steam within it. The operations going on within the cylinder consist, first, of the transfer of a portion of the entering heat into a small portion of the thickness of the walls of the cylinder, heating them to the temperature of the entering steam. This transfer of heat is most active during the period of steam admission and up to the point of cut-off. After the cut-off, the transfer continues until the lower pressure due to expansion reaches such a point that the temperature of the steam is below that of the interior surfaces last uncovered. Then the interchange of heat is reversed, and the metal gives up heat to the steam and causes what is known as re-evaporation of the particles of water on the surface of the cast-iron walls and piston. The giving up of heat from the small thickness of the surfaces which were heated during admission begins after cut-off takes place, or, to be more explicit, after the pressure begins to fall, and continues during the entire stroke. Heat is also given off from the metal of the cylinder to the steam, due to the higher temperature of the jacket, if any be used, and this occurs throughout the whole of the stroke, assuming, what is usually true, that the pressure of steam in the jacket is greater than that in the cylinder.

There is another action going on in the cylinder and that is the loss of heat due to the performance of work. There is also a loss of heat, due to radiation from those portions of the cylinder not protected by the jacket. The quantity of heat remaining after the steam has been subjected to these various complicated actions is that which passes out with the steam through the exhaust valve and is thereby "rejected," as it is termed, to the condenser or to the atmosphere. The heat analysis determines the amount of heat which is used and transferred in these various actions during the time in question, and the various percentages which these amounts bear to the total quantity of heat supplied.

APPENDIX NO. 30

TEMPERATURE-ENTROPY DIAGRAM

The study of the heat analysis is facilitated by the use of the temperature-entropy diagram, in which areas represent quantities of heat, the coordinates being the absolute temperature and entropy. Such a diagram is shown in Fig. 27*.

When the quantities given in the steam tables are plotted, two curves, *AA* and *BB*, are obtained, which may be termed the water line and the steam line, *AA* being the logarithmic curve if the specific heat

* Reproduced from Trans. Am. Soc. M. E., vol. 24, p. 758, fig. 134.

of the water is taken as constant. The diagram refers to a unit weight of the agent. The heat necessary to raise a pound of water from the temperature *ma* to the temperature *pa'* and evaporate it at that temperature is represented by the area *aa'b'qm*. If the steam be now expanded adiabatically the temperature will fall to *qs* and x per cent $= \frac{as}{ab}$ will remain as steam, the rest being liquefied. If the steam is now rejected, it carries away with it the heat *sqma*, the

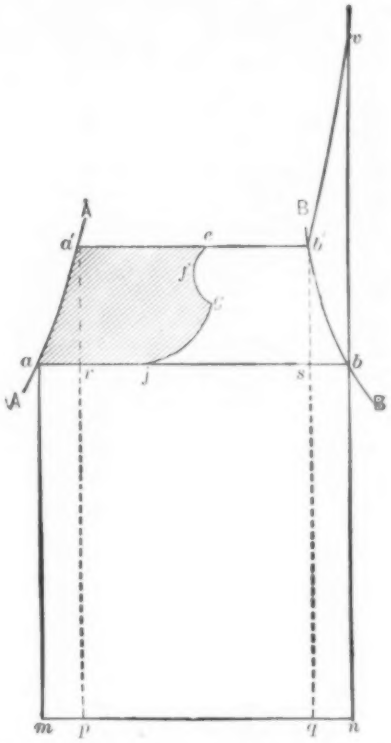


FIG. 27 TEMPERATURE-ENTROPY DIAGRAM

work area being *a'b'sa*, from which must be deducted the work w (expressed in heat units) to pump a pound of water into the boiler The efficiency of this cycle is evidently

$$\frac{h + L_1 - xL_2 - w}{h + L_1}$$

in which

$$x = \frac{ar + a'b'}{ab} = \log_e \frac{T_1}{T_2} + \frac{L_1}{T_1} + \frac{L_2}{T_2}$$

By the action of the walls a portion of the steam is liquefied prior to the expansion, which therefore begins at *e*, and since the cooling action of the walls continues, the expansion line falls off to *ef*, from which point a reverse action takes place, and the expansion line bends over to *g*. Finally, since the release takes place before the condenser temperature is reached, the heat rejection starts at *g*, following a line of equal volume until the exhaust port temperature is reached at *j*. If heat is added during expansion enough to keep the steam theoretically saturated, as, for example, by a water jacket, such additional heat is represented by the area *b'bnq*, and the additional work obtained by the triangle *b'bs*. If the steam is superheated sufficiently to give by expansion theoretically dry steam at the end,

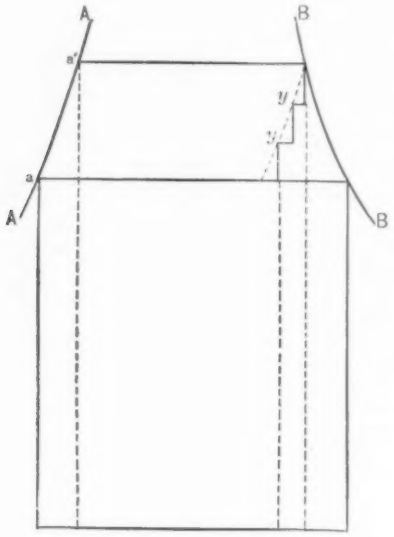


FIG. 28 TEMPERATURE-ENTROPY DIAGRAM

such additional heat is represented by the area *b'vnq* and the additional work by *b'vbs*. Neither of these extra amounts of work are realized in practice, and it is evident from the diagram that the heat thus applied is in both cases less efficient than in the principal cycle. Nevertheless the action in each case is to bring the point *e* nearer the point *b'*, and to effect a notable net economy.

The Carnot cycle would be obtained if in the Rankine cycle the rejection of heat were stopped at *r*, and the temperature of the mixture raised to *a'* by compression. This cannot be practically accomplished, but a system of feedwater heaters has been suggested and exemplified in the Nordberg engine, which is theoretically a close equivalent to it. Where steam is expanded in, say, three cylinders,

the feedwater may be successively heated from the receiver intermediate between each pair, the effect of which is illustrated in Fig. 28*. The expansion line follows the heavy line, being carried over to

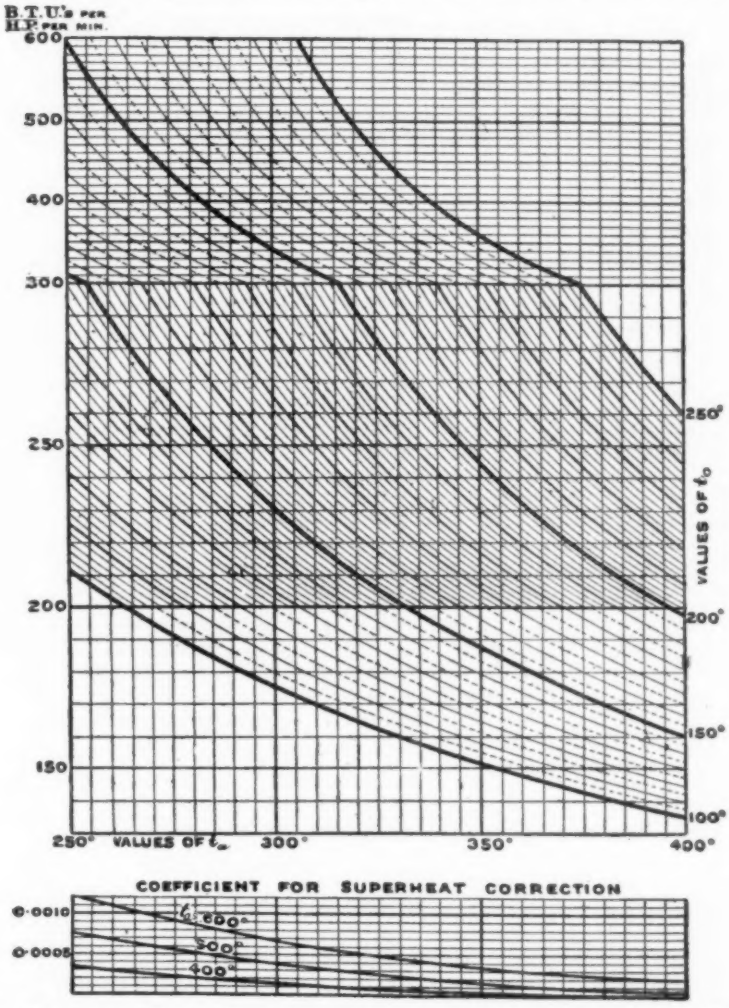


FIG. 29* HEAT UNITS EXPENDED PER I.H.P. PER MINUTE BY IDEAL STEAM ENGINE CONFORMING TO RANKINE CYCLE

y by the first feedwater heater and to y' by the second feedwater heater. With an infinite number of such feedwater heaters, the line yy' would be parallel to aa', and the cycle equivalent to that of Carnot.

* Reproduced from Trans. Am. Soc. M. E., Vol. 24, pp. 759, 761, figs. 135, 136.

APPENDIX NO. 31

RATIO OF ECONOMY OF AN ENGINE TO THAT OF AN IDEAL ENGINE

The ideal engine is assumed to be the same as that used for the British Institution of Civil Engineers Standard. It is one which conforms to the Rankine cycle, in which an ideal cylinder with no clearance is used. Steam is admitted at constant pressure and expanded adiabatically to the back pressure.

The ratio desired is obtained by dividing the heat consumption of the ideal engine by that of the actual engine.

To facilitate the determination of this ratio the heat consumption of the ideal engine may be obtained from the chart given in Fig. 29 which is copied from the Proceedings of the Institution of Civil Engineers. In this chart t_s represents the temperature of saturated steam at the boiler pressure in deg. fahr.; t_{ss} that of the steam furnished to the engine should there be superheating; t_e that of the exhaust. The heat units consumed per minute per i.h.p. can be read off directly from the curves given in the upper portion of the diagram; thus if the temperature of the exhaust (t_e) is 212 deg. fahr., and the temperature of the steam at boiler pressure is 350 deg. fahr., the heat consumption is 265 heat units per i.h.p. per minute. If the steam is superheated, the quantity is corrected by employing the factor obtained from the lower part of the diagram. Opposite the temperature of saturation corresponding to the pressure in the boiler, and on the curve corresponding to the temperature of superheated steam (t_{ss}) is found a coefficient. This coefficient multiplied by the exhaust temperature, and by the heat consumption per minute, gives the deduction to be made on account of the superheating. Thus, if the temperature of the superheated steam is 500 deg. fahr. in the case noted, we find opposite 350 deg. for t_s, and on the curve for $t_{ss} = 500$ deg., the coefficient 0.00015. The correction is $0.00015 \times 212 \times 265 = 8.5$ B.t.u., and the heat consumption with superheated steam is $265 - 8.5 = 256.5$ B.t.u. per i.h.p. per minute.

APPENDIX NO. 32

LEAKAGE TEST OF INSIDE PLUNGER PUMP

The leakage of an inside plunger is most satisfactorily determined by making the test with the cylinder head removed. A wide board or plank may be temporarily bolted to the lower part of the end of the cylinder, so as to hold back the water in the manner of a dam, and an opening made in the temporary head thus provided for the reception of an overflow pipe. The plunger is blocked at some intermediate point in the stroke (or, if this position is not practicable, at the end of the stroke), and the water from the force main is admitted at full pressure behind it. The leakage escapes through the overflow pipe, and it is collected in barrels and measured.

Should the escape of the water into the engine room be objectionable, a spout may be constructed to carry it out of the building. Where the leakage is too great to be readily measured in barrels, or where other objections arise, resort may be had to weir or orifice measurement, the weir or

orifice taking the place of the overflow pipe in the wooden head. The test should be made, if possible, with the plunger in various positions.

In a case where it is difficult to remove the cylinder head, it may be desirable to take the leakage from one of the openings provided for the inspection of the suction valves, the head remaining in place.

It is here assumed that there is a practical absence of valve leakage, a condition of things which ought to be attained in all well-constructed pumps. Examination for such leakage should be made first of all, and if it occurs and it is found to be due to disordered valves, it should be remedied before making the plunger test. Leakage of the discharge valves will be shown by water passing down into the empty cylinder at either end when they are under pressure. Leakage of the suction valves will be shown by the disappearance of water which covers them.

If valve leakage is found which cannot be remedied, the quantity of water thus lost should also be tested. The determination of the quantity which leaks through the suction valves, where there is no gate in the suction pipe, must be made by indirect means. One method is to measure the amount of water required to maintain a certain pressure in the pump cylinder when this is introduced through a pipe temporarily erected, no water being allowed to enter through the discharge valves of the pump.

APPENDIX NO. 33

LOCOMOTIVE TESTING STATIONS

The shop test described in the Locomotive Code cannot readily be made unless a testing station is available where the locomotive may be sent, and where this work may be carried on. Such a station is too expensive to be constructed for the purpose of ascertaining the performance of an individual locomotive. How to arrange and install a shop-testing plant seems therefore a matter for independent consideration.

A number of universities have these plants in use or in process of installation, a notable instance being the Purdue University, which was the first to inaugurate the shop test.

The most elaborate work that has recently been done in the shop-testing line is that which was carried on in 1904 by the Pennsylvania Railroad Company at the Louisiana Purchase Exposition, St. Louis. A full description of the plant there installed, and of the tests made upon it, may be found in a volume published in 1905 by that company, entitled "Locomotive Tests and Exhibits." A study of this book is commended to any who require detailed information regarding the design, construction, and operation of shop-testing apparatus. The plant and tests referred to were planned under the counsel of an advisory board, three of whom were members of the Society.

APPENDIX NO. 34

LOCATION OF APPARATUS AND INSTRUMENTS FOR LOCOMOTIVE TESTS

The location of apparatus and instruments in a shop test do not need to conform to restrictions of space such as pertain to road tests, and substantially the same considerations apply in that case as in the ordinary stationary steam plant. Reference may therefore be made to the Boiler and Engine Codes for any needed instructions in this matter.

As to the locations and requirements for road tests, the following directions should be observed:

(1) The water meter should be attached to the suction pipe of the injector at a point where it can be conveniently read when the train is in motion. A check valve should be provided to prevent hot water backing through it when starting and stopping the injector, as also a strainer.

(2) The indicator driving rig should be some form of pantograph motion, with a light rod transmitting the reduced motion to a point near the indicator. Lack of room and facility of operation make it desirable to use a single indicator for each cylinder, connected to the two ends by a three-way cock. It is best to carry the pipes to the side of the cylinder rather than the heads, and provide a branch leading to the steam chest, sharp bends being avoided, and the outside protected from radiation. Absolute rigidity of the indicator cock is essential, and it should be obtained by clamping it securely to the cylinder.

(3) A special steam gage should be employed which can be read by the observer stationed in the pilot box at the front end.

(4) A rod should be attached to the reverse lever and carried forward to the pilot box, where a scale is provided to show its position. A rod should likewise be connected to the throttle-valve lever for the same purpose.

(5) The draft gage may be a simple U-tube containing water, and it can best be placed in the pilot box. A rubber tube connection should be provided which may be attached at will to pipes leading to the smoke box and furnace.

(6) The flue thermometer should be one graduated to 1000 deg. fahr., and inserted so that the bulb occupies a position below the tip of the exhaust nozzles and between them and the front tube plate of the boiler.

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

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Woods, exotic 1896
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FOREIGN REVIEW

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THIS MONTH'S ARTICLES

Besides notices of several investigations made in Holland and Germany in connection with concrete and reinforced-concrete practice, there are given this month accounts of experiments with methods of treating boiler sheets to avoid initial stresses and cracking when assembling and riveting the boiler shell. Experiments upon gas velocity and energy of flow are also outlined.

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more expensive, cast-iron pipes in economizers being as good as or better than wrought-iron ones, or slag cement practically equal to portland cement. On the other hand, new apparatus sometimes results from such experiments. For example, W. Gaede, from what appeared a purely theoretical investigation of external friction between gas molecules and the walls of a vessel, and the angle of rebound of the molecules, designed a new vacuum pump, exceeding mercury vacuum pumps in speed and flexibility of operation. That manufacturers are not slow to take advantage of experiments, is shown in the installation for testing materials of the Protos Automobile Company of Nonnendamm, which includes apparatus for metallographic and microphotographic tests and the application of the latest processes such as the Heyn etching process, and which not many universities in this country or abroad can equal for completeness.

More and more space in engineering publications is being taken by new problems hardly mentioned a few years ago. Thus, whole issues last month were devoted to heavier-than-air flying machines by Der praktische Maschinen-Konstrukteur (no. 18) and Mémoires de la Société des Ingénieurs Civils de France (July), besides a large number of articles in other periodicals, one of which, by M. Rateau, from a paper read some time ago before the Association Technique Maritime, will be abstracted in the next number of THE JOURNAL. Some attempts at solving another new problem, driving an internal-combustion engine on stored oxygen under the peculiar working conditions existing in a submarine boat, are reported in this issue.

Among articles of a theoretical character, attention is called to the continuation of Karl Mayer's investigation on the theory of plunger pumps, and Professor Dwelshauvers-Dery's very favorable criticism of J. Paul Clayton's method of logarithmic indicator diagrams, presented at the Spring Meeting of the Society in Cleveland. Mr. Mayer is the author of an important work on the theory and design of plunger pumps and is considered an authority on this subject in Germany. His article, together with the Borsig tests of duplex-spring valves in pumps, of which an abstract was given in the October issue of THE JOURNAL, form a noteworthy contribution to the theory of this type of machinery.

Professor Dwelshauvers-Dery calls attention to the fact that

the usual steam engine indicator diagram made up of the head and crank diagrams does not at all represent the average conditions in the engine cylinder, and that, moreover, the law of compression of steam in the engine cylinder differs from that of expansion. It is a rather interesting fact, considering that the engineering world has been using the steam indicator, in one form or another, almost ever since the days of James Watt, that Prof. Dery, writing in the *Revue de Mécanique*, a highly technical publication, still finds it necessary to emphasize the necessity of certain precautions in using this instrument.

Air Conditioning

METHODS OF VENTILATION AND REFRIGERATION OF ELECTRIC PLANTS (*Procédés de ventilation et de réfrigération des installations électriques*, H. Marchand. *L'Industrie électrique*, vol. 21, no. 497, p. 397, September 10, 1912. 3 pp. *p*). General discussion of ventilation and methods for keeping cool electric generating stations.

Air Machinery

STEEL PLANT BLOWING ENGINES BUILT BY LEFLAIVE & CIE (*Machines soufflantes d'aciéries construites par la maison Leflaive & Cie. Le Génie Civil*, vol. 61, no. 19, p. 373, September 7, 1912. 4 pp., 10 figs. *d*). Reciprocating steam engines with two parallel cylinders, according to the author, are better adapted for driving blowers in steel plants than either steam turbines or gas engines: the blowing of a converter is an operation of only a few minutes during which the engine has to reach from zero, its maximum speed, and return to rest; the air pressure varies from 0 to 3 kg. or 42.6 lb. per sq. in., and the power developed from zero to thousands of horsepower. The reciprocating steam engine can satisfy better than any other motor the required conditions of intermittent working, rapid starting and stopping, working at constantly varying speed and output. When the blowers are driven by gas engines, the problem is somewhat simplified by having the engine work all the time, though at various rates; but this is not economic.

The described blower is provided with Hoerbiger valves. The article contains a detailed description of the Collmann steam engine with the Collmann valve gear with catch.

Fuels

FABRICATION OF LEAN GAS BY DRY DISTILLATION OF SLUDGE FROM SEPTIC TANKS (*La fabrication de gaz pauvre par distillation pyrogénée des boues de fosses septiques*, L. Cavel. *Le Génie Civil*, vol. 61, no. 21, p. 418, September 21, 1912. 2 pp. *g*). Discussion of the question of disposal of the sludge from septic tanks first by drying it by exhaust steam from the pumping machinery, and then gasifying it for power purposes. *Sludge gas* is claimed to have a high calorific value. The article does not report any actual application of this scheme.

BRIQUETTING OF COAL WITH THE ADDITION OF NAPHTHALINE (*Steinkohlenbrikettierung mit Naphtalinzusatz*, Grahn. *Glückauf*, vol. 48, no. 38, p. 1536, September 21, 1912. 1½ pp., 2 figs. *d*). The first process used in Germany of *coal briquetting with naphthaline as a binder* was that of Buss-Fohr in which naphthaline was introduced together with superheated steam having a temperature somewhat higher than the boiling point of the naphthaline, the idea being that the naphthaline would evaporate, and thoroughly permeate the coal mass. The temperature of the steam proved, however, to be too low, and no higher temperature could be used owing to the difficulty of handling in the presses masses that are too hot.

The Shüring process described in the present article avoids this difficulty first by evaporating the naphthaline in a special gasifier, and then introducing it by steam pressure into the deeper zones of the coal dust. Tar is still used as a binder, but instead of 6½ per cent required when no naphthaline is added, good briquettes can be produced with 4½ to 5 per cent tar and only 0.3 per cent naphthaline.

MEASUREMENT OF COAL CONSUMPTION IN REVOLVING TUBULAR KILNS BY FLUE GAS ANALYSIS (*Messung des Kohlenverbrauches von Drehröfen durch Rauchgasanalyse. Tonindustrie-Zeitung*, vol. 36, no. 106, p. 1147, September 7, 1912. 3 pp. *pt*). Account of a paper presented by Dr. H. Kühl at the last annual meeting of the Society of German Portland Cement Manufacturers on the determination of *coal consumption of revolving tubular kilns*. The determination of coal consumption by weighing the coal and clinker is unreliable, because the fuel has to be measured before entering the furnace, while the clinker is measured after leaving the furnace and rotary coolers. There is therefore a considerable period between the two measurements, and to obtain any reliable results the duration of the test must be so long as to make the time difference between the measurements comparatively insignificant. Each test would then be conducted for several days, during which a perfect uniformity of operation must be maintained, a thing manifestly difficult and inconvenient. A better way of estimating the consumption of coal has been found in flue gas analysis.

The principle of the method consists in analyzing the mixtures of gases produced by the simultaneous process of combustion of a definite amount of coal and disoxidation of a certain amount of raw material, and calculating from this analysis the ratio of the two processes, i. e. how much coal was required for the disoxidation of a given amount of raw material, or burning of a given amount of clinker. The following formula is deduced:

$$x = \frac{K_1 V}{K_2 W - K_3 V}$$

where x is the number of *kg* of coal required to produce 100 kg of ash-containing clinker, such as leaves the kiln; $K_1 = \frac{400_m}{100 - n}$ where m is the contents in per cent of $CaCO_3$ in the raw material, and n loss due to

burning; $K_3 = \frac{b}{3} + c + \frac{ad}{7} - \frac{e}{8} + \frac{f}{8}$, where b is the content of carbon in the coal, c that of hydrogen, d of nitrogen, e of oxygen, and f of sulphur, all in per cent; $a = 0.262$, the volumetric ratio of oxygen to nitrogen in air; $K_2 = \frac{b}{3} - \frac{mg}{25\ (100-n)}$, where g is the content of ash in coal in per cent; and finally $V = {}_a s - q + \frac{t}{2}$, and $W = p + t$, where p,q,s,t represent the contents, in per cent, in the flue gases of carbon dioxide, oxygen, nitrogen, and carbon monoxide respectively.

The following example is used to show the simplicity of this method of determination. From the analysis of the raw material and coal the following values have been obtained for the constants: $K_1 = 5200$, $K_2 = 315$, $K_3 = 272$. Hence

$$x = \frac{5200V}{315W - 272V}$$

From the gas analysis it was found in a special case that $p = 27.6$, $q = 0.2$, $t = 0.2$, and $s = 72.0$, all in per cent. Hence:

$$V = 0.262 \cdot 72.0 - 0.2 + 0.1 = 18.8$$
$$W = 27.6 + 0.2 = 27.8$$

Substituting these values in the formula for x, the following is obtained:

$$x = \frac{5200 \cdot 18.8}{315 \cdot 27.8 - 272 \cdot 18.8} = 26.8 \text{ kg per 100 kg clinker}$$

Since in regular kiln operation the contents of carbon monoxide in the flue gases are negligible, t may be taken as equal to zero. This not only considerably simplifies the formulae, but allows the use of the usual self-registering apparatus for the oxygen and carbon dioxide contents, which, together with the simple analysis of coal, afford an easy and convenient method of controlling the coal consumption of rotary kilns. A full derivation of the above formulae is given in the article.

Hydraulics

BLUE COAL (*La Houille bleue*, S. F. *La Lumière électrique*, vol. 19, no. 37, p. 340, September 14, 1912, 4 pp. *t*). The French call waterpower White Coal, and Jules Severin, in his paper before the Dijon meeting of the French Association for the Advancement of Sciences, called by analogy the *force of the tides* Blue Coal. The author proposes to construct a reservoir of about 1 km sq., and 7 to 14 m (23 to 46 ft.) deep, to take in water at high tide, and deliver it to water turbines later; the power is to be utilized according to the Taylor system used at the Rockland plant.

SHUTTING-OFF DEVICES ACTUATED BY THE EXPANSION OF LIQUIDS (*Absperrorgane durch den Auftrieb von Flüssigkeiten betätigt*, J. Fichtl, *Gesundheitsingenieur*, vol. 35, no. 36, p. 701, September 7, 1912. 6 pp., 13 figs. *dg*). General discussion of the principles of action of shutting-off devices, and description of some German types. The author investigates the mutual relations between the form of the float, its size and closing time and establishes, among other things, that the relation between the time and amount of liquid flowing out is represented not by a straight line, but by a logarithmic curve.

To the Determination of Pulsations in the Forebay (*Zur Ermittlung der Schwingungen im Wasserschloss*, Ph. Forchheimer. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 32, August 10, 1912. 1½ pp. *t*). The author shows how the pulsations in a forebay may be determined without the usual limiting assumptions, such as the proportionality of pressure loss to velocity, or the use of small final differences instead of differentials. He deduces the following equation:

$$mZ+1-\ln(mZ+1)=mh+1$$

for the application of which the following data must be known: velocity of water in the state of resistance u_0, cross-section of penstock F, water area F_1, length of penstock l, loss of pressure head in the penstock $\frac{lu^2}{c^2R}$, then $m=2gF_1:c^2RF$; h is the loss of pressure head in the penstock in the state of resistance. A numerical example is given calculated by the author's formula and, for the sake of comparison, by the method of final differences.

Internal Combustion Engines

Electric Power Plants in Mine and Steel Works in Rhineland-Westphalia, Belgium, Northern France and England (*Elektrische Kraftanlagen in Rheinland-Westfalen, Belgien, Nord-Frankreich and England*, K. Hoefer. *Zeits. für des gesamte Turbinenwesen*, vol. 9, nos. 22 to 24, August 10, 20, and 30, 1912. 10 pp., 20 figs. The same article in a somewhat abridged form in *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 32, p. 1281, August 10, 1912. 10 pp., 25 figs. *d*). Description of *power transmission and prime movers* in the electric power plants in mine and steel works in the above named countries, with the description of some constructive details (mainly in *Zeits. für Turbinenwesen*). The article contains a table showing the *reserve power in power plants* in various countries. On the average about a third of the power of the machinery is held in reserve in case of accident, this reserve being practically equal for steam and *gas engines*, which shows that the latter are considered fully *reliable* in operation. The *smallest reserve* is kept in the case of *steam turbines*.

Tests of a Jahns Annular Producer at the Mine Hohenzollern (*Versuche an dem Ringgenerator-System Jahns auf der Hohenzollerngrube. Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 37, p. 385, September 13, 1912. 3 pp., 5 figs. *de*). In the *Jahns producer* four chambers are built in a ring, with a common vertical passage along the axis. This passage is connected with the chambers above by openings provided with closing doors, and with those below by a circular duct. The purpose of combining the four chambers into a single ring is to conduct the gases (products of incomplete combustion), from a freshly charged chamber into the other chambers which have been in operation longer and so had time to become hot. The gases rich in coal-tar are forced, together with a steam jet, under the glowing fuel bed, and are thus transformed into permanent gases free from tar. The article contains a de-

tailed description of operation and full data of tests, the gas produced being used for heating steam boilers.

Gas Engine-Dynamo Units (*Gasdynamos. Zeits. für Dampfkessel und Maschinenbetrieb*, vol 35, no. 38, p. 395, September 20, 1912. 3 pp., 6 figs. *dp*). Description of some German gas engine-dynamo direct-coupled units, with some suggestions as to the design of the gas engines in such units.

The Valveless Motor and Gear of the Fischer Car, Zurich (*Der ventillose Motor und das Getriebe des Fischer-Wagens, Zürich. Auto-Technik*, vol. 1, no. 17, appended to *Allgemeine Automobil-Technik*, vol. 3, no. 37, September 13, 1912. 3 pp., 8 figs. *d*). *The Fischer slide-valve motor* belongs to the same class as the Mustad motor described in The Journal, March 1912, p. 414, in that it has separate valves for the admission and exhaust, each valve having the width of about a quarter of the cylinder circumference, and both being exactly similar and therefore interchangeable. Only half of the piston surface is covered by the valves, the other half, or rather the other two opposite quarters, lying against the water-cooled cylinder walls. The piston is provided with three piston rings which by their spring action help to obtain a tight fit between the valves and the outer wall; a fourth, stationary ring in the cylinder casing provides packing for the part where the piston does not reach. The article describes also the method of machining the valves, and the *change speed gear* of the car, which, according to the author, is suitable only for small and cheap cars.

Gas Producers for the Gasification of Liquid Fuel (*Gaserzeuger für Vergasung flüssiger Brennstoffe*, Gwosdz, *Braunkohle*, vol. 11, no. 23, p. 357, September 6, 1912. 10 pp., 11 figs. *d*). In the Rincker and Wolter process used in Watergrasmeer, near Amsterdam, Holland, for producing gas from tar, the liquid fuel is gasified by passing it through a layer of glowing coke, without the admission of steam. The producer consists of two shafts connected in parallel. The coke is heated to glowing first in one shaft, into which tar is then introduced to be gasified. It is kept there until the temperature reaches a certain limit when the gases are led out to be used. The connections are changed in such a way that the air blast, in passing through the first shaft, drives the gases, only partly decomposed owing to the low temperature, into the second shaft where they pass through the glowing fuel from top downwards and are converted into stable gases.

This process has, however, the following disadvantages: (*a*) the air blast is passed through one of the shafts twice, and in the meanwhile the shaft has time to cool down thus retarding the decomposition of the oil vapors; (*b*) during the blowing of the producer the gases escape and the oil vapor contaminates the atmosphere.

In the improved type of the Berlin-Anhalt Machine Construction Company both shafts are connected in series, and get the hot blast alternately, the one which the hot air current strikes first being connected with the apparatus using the gas. The blowing of the producer shafts in series permits of keeping both generators hot, while alternating the direction of blowing has the advantage of the shaft which is blown first being

hot from the previous blowing. Both generators are therefore better heated and the decomposition of oil vapors proceeds more effectively. The article describes also the Pintsch, Blaugas, and several American types of producers for liquid fuel (E. C. Jones, Frost, Ammet, Hydrocarbon Converter Company).

KOPENHAGEN FISHING EXHIBITION OF 1912 (*Kopenhagener Fischerei-Ausstellung 1912*, Gbm. *Die Gasmotorentechnik*, vol. 12, no. 6, p. 100, September 1912. 2 pp., 4 figs. *d*). Short descriptions of some *Scandinavian motors*. The motor of the *Bergsunds* Mek. Werkstads Aktiebolag in Stockholm has the *gasifier* as shown in Fig. 1, with a number of consecutively arranged narrow metal strips *H* connected with each other and with the lower side of the cover *I*. The fuel is thrown against these strips from the nozzle *G* as shown by the dotted lines, so that each strip receives and distributes in various directions an equal amount of fuel per unit of its area, this fuel being at the same time evaporated owing

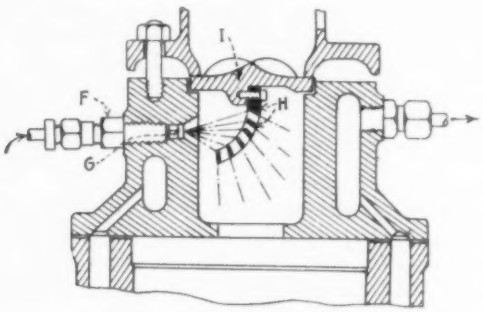

FIG. 1 GASIFIER OF THE BERGSUNDS MOTOR

to the high temperature of the metal strips. This happens just before the piston reaches its highest position, the heat of compression together with that of the metal strips igniting the fuel mixture immediately after its admission. This effective system of gasification permits of admitting the fuel into the combustion chamber so late, i. e. so near the uppermost position of the piston, that all possibility of pre-ignition is eliminated, while the strips *H* are practically at constant temperature, independent of the load, owing to the cooling produced by the fuel being injected against them. The article also describes Tuxham and Junkers motors. (For other motors at this Exhibition, see *The Gas Engine*, vol. 14, no. 10, 1912, p. 541).

GAS VELOCITY AND ENERGY OF FLOW (*Gasgeschwindigkeit und Strömungsenergie*, v. Löw, *Zeits. des Mitteleuropäischen Motorwagen-Vereins*, vol. 11, no. 17, p. 373, Mid-September 1912. 2 pp., 3 figs. *cp*). There are as yet no generally accepted rules for the *proportions of automobile valves and gas velocities*. Adler makes the suction and exhaust valve cross-sections equal, and uses 102 m (332 ft.) per sec. as the velocity of flow when the maximum velocity of the car is 125 km (77 miles) per hour.

Benz has the cross-section of the suction valve nearly twice as large as that of the exhaust valve, and the velocity of flow 45 m (147 ft.) per sec. for the maximum velocity of the car 134 km (82 miles) per hour. Horch uses a ratio of suction valve and exhaust valve cross-sections as 4 to 3, and for the inlet velocity of the charge 57 m (187 ft.) per sec. at the car speed of 87 km (54 miles). Although the most advantageous ratio of valve cross-sections can be found only by suitable tests, the author tries to establish certain facts in this connection by means of the diagram in Fig. 2 A., where *aa* is the atmospheric line, *vv* absolute zero or vacuum line, and the pressure process during the greatest part of the exhaust stroke is represented by the line 1-2. At the point 2 the piston is supposed to have reached the end of the exhaust stroke and to be in the neutral position, the pressure being not atmospheric, but above it, 2-3, owing to throttling in the exhaust outlet. When the piston starts on a new stroke,

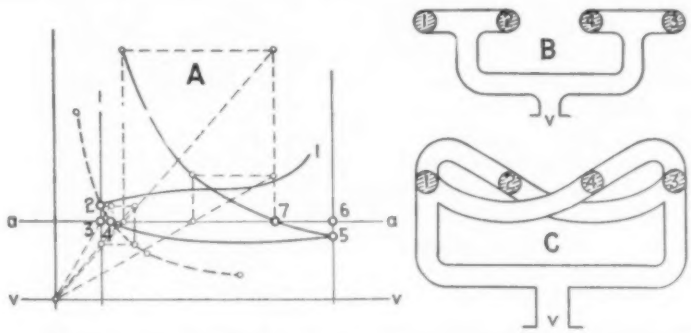

FIG. 2(A) PRESSURE DIAGRAM FOR THE EXHAUST AND SUCTION STROKE OF AN AUTOMOBILE ENGINE (B) AND (C) SUCTION PIPING CONNECTIONS IN A MULTI-CYLINDER AUTOMOBILE ENGINE

there is an expansion, but the atmospheric pressure is reached only at 4; it is therefore only from there on that the real suction of the piston begins, 3-4 representing the volumetric cylinder loss at the beginning of the suction stroke. This cannot be helped by opening the suction valve when the piston is in its neutral position because then the expansion in the suction piping forces back the fresh mixture which in that case does not begin to reach the cylinder until the piston is at 4. The throttling pressure 2-3 must therefore produce always the volumetric cylinder loss 3-4.

Should the throttling at the end of the suction stroke be equal to that at the end of the exhaust stroke, or 5-6 be equal to 2-3, then, since the compression curve rises much slower than the expansion curve falls from 2 on, the volumetric loss 6-7 would be about six times as large as that at the beginning of the suction stroke, or 3-4, which shows that with equal throttling areas at the suction and exhaust valves the volumetric cylinder loss produced by the throttling at the suction valve is considerably greater than that produced by the throttling at the exhaust valve. The suction valve ought therefore to be larger than the exhaust valve.

It is, however, worth noticing that the Daimler Motor Company who were among the leading builders of engines with unequal valves, have now equal valves in their aeronautic motors.

It is usually accepted that the volumetric cylinder loss due to throttling at the valves may be reduced by late closing of the valves. Leading constructors are, however, by no means agreed as to just how late the closing should be, and practice varies between 0 and 49 deg. for the suction valve and 0 and 22 deg. for the exhaust valve. In multi-cylinder engines, owing to the energy of flow due to high gas velocities, it may be even advisable to select different closing times for the valves of different cylinders. Cylinder 1 in Fig. 2B must set in motion the gas in the left part of the suction piping, while cylinder 2 finds it already in motion, and therefore receives more gas with the same valve gear. The same happens with the cylinders 3 and 4 for the right side of the suction piping. A more uniform system of feed is represented when the suction piping is arranged as shown in Fig 2C: when there is suction in cylinder 1, it receives the larger part of its gas through the left side of the piping from the carbureter *c*; but there is an additional stream of gas coming past 4, after the closing of its valve after the completion of its previous suction stroke. When the suction stroke in 1 is ended, the stream of gas continues to flow first to 2, and then to 3 the last of which also receives some mixture direct from the carbureter, through the right side of the piping, this stream continuing to 4 after the end of the suction stroke in 3. The flow is therefore continuous and uniform, and there are no particular losses of the energy of flow. The author insists, however, on the necessity of a systematic series of experiments for the full elucidation of all these points.

On the Influence of Sulphur in Liquid Fuels used in Motors (*Uber den Einfluss des Schwefels in flüssigen Brennstoffen beim Motorenbetrieb*, E. Gräfe. *Der Ölmotor*, no. 2, 1912, through *Braunkohle*, vol. 11, no. 24, p. 383, September 13, 1912. *p*). In Diesel and other internal-combustion engines oils, mostly by-products of the coal-tar industry, are used containing up to 2 per cent of *sulphur*, and the question arises in how far this may *harm the engine*. The products of oxidation of sulphur, viz. sulphur dioxide and sulphur trioxide, do not attack iron unless they are dissolved in water, and form sulphurous or sulphuric acid. In the motor cylinder or in a boiler furnace where water is present only in the form of steam, there can be no corrosion due to the presence of sulphur, but wherever there is condensation of water, as in the exhaust piping, the iron is strongly attacked. This may be prevented, however, either by making the exhaust piping short, or by covering it with lead. On the whole, for the motor itself, the presence of sulphur is therefore practically harmless. Cp. *Fuel Oil Journal*, October 1912. p. 15.

New Working Processes for Internal-Combustion Engines in Submarines (*Neue Arbeitsverfahren für Verbrennungskraftmaschinen in Unterseebooten*, Dr. G. *Petroleum*, vol. 7, no. 22, p. 1249, August 21, 1912. 2 pp., 3 figs. *d*). Description of some new *submarine engines* and discussion of their *working processes*. Submarine engines during the run under

water are sometimes fed by pure oxygen, the exhaust gases being used over again either in their original state, or without their carbon dioxide, i e. with practically only their nitrogen. Contrary to this practice, Dr. Jaubert

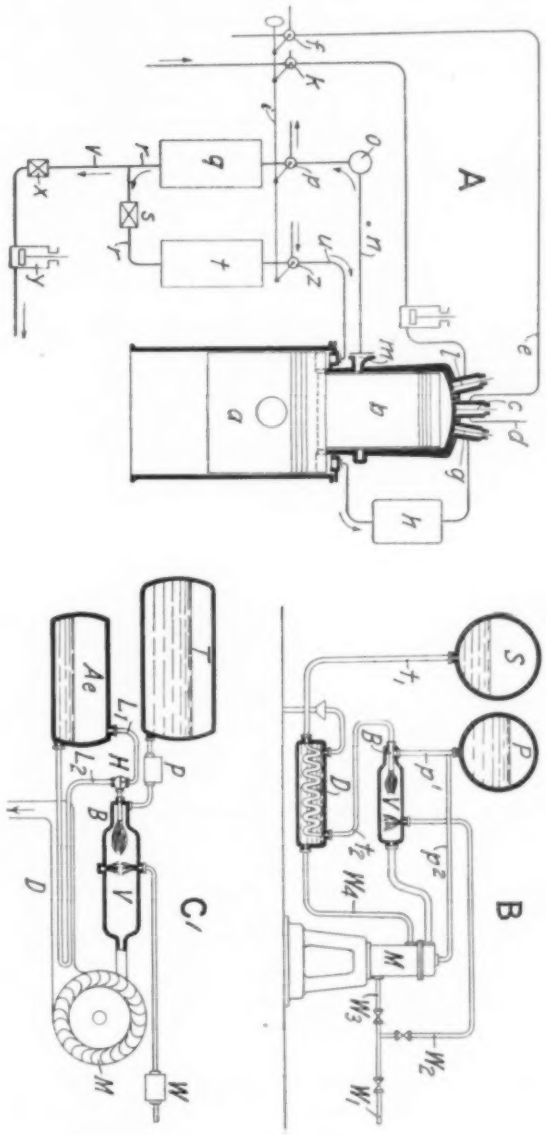

FIG. 3 SUBMARINE ENGINES WORKING ON STORED OXYGEN OF GASES RICH IN OXYGEN

of Paris recommends the elimination of all nitrogen from the exhaust gases and the use of only carbon dioxide, on the assumption that concentrated

carbon dioxide in the absence of water vapor will take part in combustion in accordance with the following formula:

$$C_7H_{16}+4CO_2+Heat=5\frac{1}{2}C_2+8H_2O+Heat \quad [1]$$

with the final products (coal and water) such that they can be conveniently thrown overboard. The engine is represented schematically in Fig. 3A. The pistons of the two cylinders which lie one above the other have a common connecting rod. Cylinder *a* acts as a compression pump and delivers the gas to *h*; the working cylinder *b* is provided with the valve *c* for the admission of petroleum coming through the pipe *d*, and injected under gas pressure (air or carbon dioxide), the compressed gas being brought to it by the pipe *e* provided with a three-way cock *f*. The pump *j* driven by the engine may be set in connection, by means of the three-way cock *k*, either with the atmosphere or the oxygen tank. The exhaust pipe *m* of the cylinder *b* communicates by the pipe *u* with collector tank *o* which in its turn may be connected by means of the three-way cock *p*, either with the atmosphere or with the compressor and separator *g*, the latter opening through the pipe *r* and pressure regulator *s* into the cleaner *t*. This cleaner is filled with peat or a similar mass, and by the three-way cock *z* may be set in communication either with the suction opening of the cylinder *a*, through pipe *u*, or with the atmosphere. The cocks *z,p,k,f* can be simultaneously regulated by the rod *i*. During the run above the surface the cocks *k,p*, and *z* are connected with the atmosphere, and the petroleum injection cock *f* with the compressed air tank. The piston *a* takes in air and delivers it to *h* whence it is taken for scavenging the cylinder *b*, while the pump *j* supplies atmospheric air for the explosive mixture to the same cylinder. During the underwater run the petroleum is injected by compressed carbon dioxide coming through the pipe *c*, the same gas being used also for scavenging. The exhaust gases are taken to the compressor and separator *g* in which the water vapor is condensed, and coal particles and unconsumed oil retained, while the clean carbon dioxide is sent on to the pipe *u* and cylinder *a*, and is later used for scavenging, as well as for the injection of fuel. Oxygen is introduced into cylinder *b* by the pipe *l*. Since no external air is introduced, the residual nitrogen is gradually expelled, and the exhaust gases contain practically only carbon dioxide and some oxygen. Simple combustion follows the equation

$$C_7H_{16}+11\,O_2=7\,CO_2+8\,H_2O \quad [2]$$

and this, together with formula [1] gives the equation for the total reaction

$$2\,C_7H_{16}+11\,O_2+4\,CO_2=7\,CO_2+5\frac{1}{2}\,C_2+16\,H_2O \quad [3]$$

In this process the residues of combustion contain a large amount of carbon dioxide thus considerably reducing the amount of this gas which would otherwise have to be stored on board the vessel, and only half the oxygen required according to equation [2] is used when the process thus takes place in accordance with equation [3]. A further advantage consists in the exhaust being only carbon dioxide soluble in water, and therefore attracting no attention of the enemy, as do air bubbles.

Another arrangement of submarine engines is shown in Fig. 3B, in which *nitrogen tetroxide* is used as the oxygen carrier. Since, however,

both petroleum and nitrogen tetroxide are liquids, one of them must be evaporated before reaching the cylinder in order to produce a good atomization of the mixture. Nitrogen tetroxide, and not petroleum, is evaporated because it possesses a lower boiling point, so that cooling water which leaves the cylinder jacket at about 50 deg. cent. (122 deg. fahr.) can be used for this purpose (exhaust gases are not used since the necessary apparatus would take more room than can be given to it on board a submarine). *P* is the petroleum tank, *S* contains nitrogen tetroxide. The combustion chamber *V* is provided with the burner *B* which receives a small amount of petroleum through the pipe p_1, and vaporized nitrogen tetroxide throughout the pipe t_2, there occurring a partial combustion and decomposition of the nitrogen tetroxide which results in the production of a gaseous mixture rich in oxygen, while water is injected into the combustion chamber through the pipes W_1 and W_2, to reduce the temperature of the oxygen-carrying mixture. The comparatively cool mixture is then introduced into the cylinder of the motor *M* where it mixes with the petroleum brought by the pipe p_2. Pipes W_1 and W_3 bring the cooling water which after passing through the cylinder jacket continues to flow through the pipe W_4 to the vaporizer *D* where the nitrogen tetroxide is evaporated.

In another type the same inventor evaporates the fuel (benzine, benzole, etc.) by the heat of exhaust gases and uses as a source of oxygen *tetranitromethan* contained at atmospheric pressure in the tank *T*, Fig. 3C and brought by the pump *P* to the burner *B*. Gaseous ethane which has a pressure of 46 atmospheres even at the temperature of 4 deg. cent. (39.2 deg. fahr.), is contained in the tank *Ac*, and is brought to the burner by the three-way cock *H* and pipe L_1. The gases of combustion in the chamber F are so much cooled by the injection of water by pump *W* that they may be used to drive the *turbine M* from which the exhaust passes through the vaporizer *D* through which also passes the pipe L_2 from the ethane tank; after the engine started, the cock *H* is set so as to connect *Ac* and *B* by the pipe L_2.

Machine Shop

Spring Hammer (*Marteau à ressorts. Portefeuille économique des machines*, ser. 5, vol. 10, no. 681, p. 131, September 1912. 2 pp., 5 figs. *d*). Description of the Ajax spring hammer. Cp. The Journal, January 1912, p. 123.

New Dust Suction Device for Polishing Machines (*Neue Staubsaugevorrichtung für Schleifmaschinen. Giesserei-Zeitung*, vol. 9, 17, p. 548, September 1, 1912. 1 p., 2 figs. *d*). The designer's problem was to construct a dust suction apparatus which might be used in connection with polishing machines in which the polishing wheel does not remain in one position, but travels around the piece of work. Such a device, patented in Germany by the firm Frederich Schmaltz, in Offenbach-A-M., is schematically shown in Fig. 4. The suction device travels with the wheel around the work. The dust-catcher *b* is then in constant position as regards the polishing wheel *a*, while the suction piping *c*, attached to the dust catcher passes through the filter *d* and through the center of the rotary motion of

the tool holder *e*, and finally to the bore in the hollow shaft *f*. The suitably shaped pipe *c* is fixed on the adjustable bracket *g* and therefore follows the motions of the tool holder. The other end of the hollow shaft of the polishing machine is air tight and rotatively connected with the suction piping *h*. Cp. description of a similar British device in Dust Collecting Device for Grinding Machines, *The Engineer* (London), September 13, 1912, p. 287.

Mechanics and Machine Design

BENDING OF CURVED TUBES (*Die Biegung krummer Rohre*, H. Lorenz. *Dinglers polytechnisches Journal*, vol. 327, no. 37, p. 577, September 14, 1912. 4½ pp., 5 figs. *mt*). The author derives a new simple formula for the *variation of the curvature of a pipe* through an external moment, from

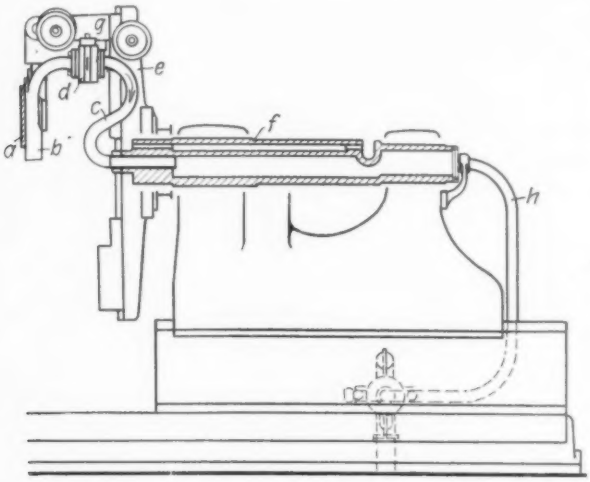

FIG. 4 DUST SUCTION DEVICE FOR POLISHING MACHINES

the conditions of equilibrium of an element of the wall of a curved tube under the assumption of conservation of double symmetry of the tube cross-section. He investigates also the direction of the moment producing increase or decrease of the radius of curvature of the tube, and derives a simple formula for the total contraction of the circular cross-section of the tube in the radial or axial direction. The problem is treated by purely mathematical methods.

ON LUBRICATION OF MACHINE PARTS (*Uber die Schmierung von Maschinenteilen*, L. Singer, *Petroleum*, vol. 7, no. 23, p. 1307, September 4, 1912. 12 pp., 64 figs. *d*). Discussion of the *theory of lubrication* mainly on the basis of Ubbelohde articles (see THE JOURNAL, June and August 1912, pp. 963 and 1245), and of the *design of machine parts subject to friction*, e.g. journals, bearings, etc., so as to obtain the best lubricating efficiency, as well as the apparatus for combined oil-and-graphite lubrication.

EXPERIMENTS ON THE DETERMINATION OF THE TORSION OF BARS WITH RECTANGULAR CROSS-SECTION, AND ELONGATION AND CONTRACTION OF AREA OF BARS UNDER TENSION (*Versuche über die Verdrehung von Stäben mit rechteckigem Querschnitt und zur Ermittlung der Längs- und Querdehnung auf Zug beanspruchter Stäbe*, O. Bretschneider, *Mitteilungen über Forschungsarbeiten*, no. 21, p. 1. 1912. 37 pp., 17 figs. *c*). Full reports of the experiments reported in THE JOURNAL, May 1912, p. 805, from a preliminary publication.

STAGING RATIO IN SPEED-CHANGING GEARS (*Abstufungsverhältnisse von Wechselgetrieben*, Fr. v. Löw. *Zeits. des Mitteleuropäischen Motorwagen-Vereins*, vol. 11, no. 16, p. 349, End August 1912. 4 pp., 4 figs. *ct*). Discussion of the problem of staging ratios, that is, whether the first transmission should be equal exactly to a quarter of the transmission when directly coupled, or more or less, and how the intervals between single stages should be proportioned, these being, according to the author, questions on which constructors are not yet agreed. In the modern cars (only German types are considered) with three-stage speed changing gears the author found only one car with the ratio between the highest and lowest transmission greater than 4, while in nearly all cars with four-stage speed changing gears this ratio was more than 4. The lowest transmission was for hill climbing, the highest for level road work. Different demands should be made in these respects regarding three and four-stage cars, particularly as the high-power car is better able to take care of variable road conditions. According to the author there is no reason why three-stage speed changing gears should be used any more since the disappearance of the belt transmission has left their construction without any real justification. The second part of the article gives data from the author's experience with the speed-changing gears of German cars under road conditions, and discussion regarding them.

RESONANCE PHENOMENA IN SUCTION PIPING OF COMPRESSORS AND GAS ENGINES (*Phénomènes de résonance dans les conduites d'aspiration des compresseurs et des moteurs à gaz*, P. Voissel. Revue de mécanique, vol. 31, no. 2, p. 155, August 31, 1912. 7 pp., 11 figs. *ct*). French translation of the article in the *Zeits. des Vereines deutscher Ingenieure* fully reported in THE JOURNAL, August 1912, p. 1248.

A NEW AXIAL COMPRESSION FORMULA (*En ny Knackningsformel*, Torbjørn Strand. *Teknisk Ukeblad*, vol. 59, no. 36, p. 460, September 6, 1912. 2 pp., 3 figs. *t*). Starting from the Schwartz-Rankine and Euler formulae for axial stresses in columns, the author by a rather complicated mathematical process derives the following *formula for axial stress*:

$$\beta_k = \frac{K}{1+\alpha\left(\frac{l}{i}\right)^2+\beta\left(\frac{l}{i}\right)^4+\gamma\left(\frac{l}{i}\right)^6}$$

where K is the compression strength of the material, $\frac{l}{i}$ slenderness ratio, α, β, γ, are defined by equations of their own, but are for each given material, functions of compression strength and modulus of elasticity.

The article also contains a table showing the relation between the slenderness ratio and axial stress.

SHOCK ABSORBERS AND ELASTIC SHACKLES (*Les Amortisseurs et jumelles élastiques. La pratique automobile*, vol. 8, no. 173, p. 3683, September 10, 1912. 2 pp., 12 figs. *d*). Description of the main French types of shock absorbers and elastic shackles.

REDUCTION GEAR FOR COAXIAL SHAFTS (*Reduziergetriebe für gleichachsige Wellen*, R. Kommer., *Motorwagen*, vol. 15, no. 26, p. 618, September 20, 1912, 3 pp., 5 figs. *d*). Description of the *Paul H. Peters epicyclic gear* consisting of two planet wheels each of which is free to make eccentric motions without changing thereby the mutual position of the teeth of the two wheels, one of these wheels engaging a fixed toothwheel coaxial with the driving shaft, and the other a wheel sitting on the

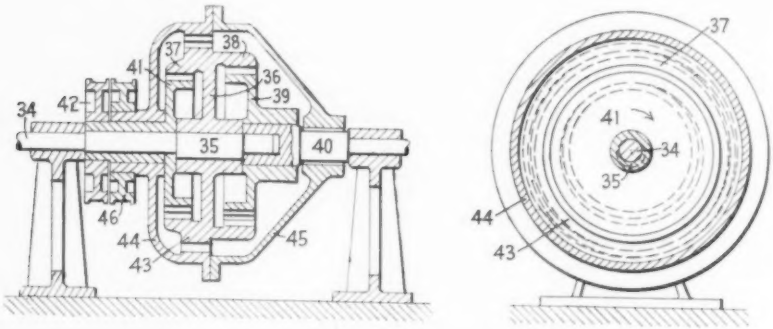

FIG. 5 PETERS EPICYCLIC REDUCTION GEAR

driven shaft. With such an arrangement the gear ratio is determined by two members of which each may be either negative or positive, according to the ratio of the teeth numbers in the separate pairs of wheels, so that the sum of the two may be equal to zero without the number of teeth of the engaging wheels being equal to each other. Ratios of transmission from 2:1 to 20000:1 and more may be obtained, either in the same direction as the driving shaft or opposite to it, without varying the direction of the driving force. The driving and the driven axes are coaxial. At least a third of all the teeth are always in engagement, this permitting the use of small teeth and a small wheel even for the transmission of large forces. It is claimed that even with the highest ratios of transmission the friction loss is below 6 per cent. Fig. 5 represents a type in which the driven shaft 40 may be driven in any direction or stopped altogether. On the middle part 34 of the driving shaft 35 sits loosely the wheel body 36 provided with three different rims 37, 38 and 43* with internal or external toothing. In engagement with the wheel rim 38 is the toothed wheel 39 sitting on the shaft 40, the latter concentrically embracing the end journal of shaft 34. The second rim 37 is in engagement

* The original has 41, but that is evidently a mistake.—EDITOR.

with the toothed wheel 41 of which the long hub sleeve sits loosely on the shaft 34 provided with a brake pulley 42. To the third rim wheel 43 of the wheel body 36 corresponds the internal toothing of the casing member 44 which with its sleeve sits loosely on the sleeve of the toothed wheel 41 while another casing member 45, bolted together with 44, loosely encloses by its sleeve the shaft 40. The sleeve of the casing member is also provided with a friction pulley placed near that of 42. Suppose now that the brake pulley 42 is held fixed, and shaft 34 rotated in the direction of the arrow. Then the internal toothed rim 37 of the wheel body 36 sitting loosely on the eccentric journal 35, rolls along the wheel 41 held by the brake; the motion of the wheel 36 retarded thereby is transmitted through its toothed rim 38 to the toothed rim of the wheel 39, the sleeve of which is wedged fast to the shaft 40. The wheels 43 and 44 rotate in the same sense as the driving shaft. If on the other hand it is the pulley 46 that is braked, the rotation of the shaft 34 in the sense of the arrow produces an engagement of 43 with the now stationary internal toothing of the casing member 44, in a sense contrary to the direction of the arrow, the rotation of the wheel body 36 being as before transmitted to the shaft 40 through the toothed wheels 38 and 39, but in a direction opposite to that of 34. When both pulleys, 42 and 46, are loose, the whole gear rotates on 39, and with the slightest load on it the shaft 40 remains stationary. All the toothed wheels are in engagement all the time, but those which are not required to do any work are allowed to run loose.

The rest of the article contains rules for calculating the number of teeth in the various wheels, and a description of another type of gear based on the same principle, but said to be specially adapted for use in turbines, automobiles, and motor boats.

Pumps

CONTRIBUTIONS TO THE THEORY OF PLUNGER PUMPS (*Beiträge zur Theorie der Kolbenpumpen*, Karl Mayer, *Die Fördertechnik*, vol. 5, no. 9, p. 202, September 1912. 4 pp., 3 figs. *mt*). Continuation of the article abstracted in THE JOURNAL, October 1912, p. 1574, containing an investigation of the case when the *angle of coupling of a plunger pump is less than 90 deg.* Since the method of the author has been made clear in the preceding abstract, only his conclusions are reported here. By means of the combined process the coefficients of cyclic variation may be made equal to each other in six out of the eight intervals occurring in each revolution, and the maximum velocities in three out of four phases per revolution. The improvement, in per cent, in the coefficient of cyclic variation due to the use of the combined process (18.16) is greater than when the method of the most suitable angle of coupling (15.67) or that of the most advantageous ratio of cross-sections (10.52) are applied singly. In addition to that the combined process permits of freely choosing the ratio of cross-sections.

Each way of selecting the angle of coupling, i.e. greater or less than 90 deg., has its advantages. With the angle of coupling greater than 90

deg. the maxima may be obtained in all four phases, and acceleration pressure may be made equal in six out of the eight intervals per revolution, but the same value may be obtained for only four out of the eight intervals of coefficients of cyclic variations occurring per revolution. Six out of those eight intervals may be made equal, however, with the angle of coupling less than 90 deg., while equal maxima of velocity may be obtained only in three phases, and only four equal maxima of acceleration resistance. The angle of coupling has therefore to be chosen separately for each particular case in accordance with the purpose of the pumping installation. The article is to be continued.

EXTERNAL FRICTION OF GASES AND A NEW PRINCIPLE FOR AIR PUMPS: MOLECULAR AIR PUMP (*Die äussere Reibung der Gase und ein neues Prinzip für Luftpumpen: Die Molekularluftpumpe*, W. Gaede. *Physikalische Zeits.*, vol. 13, no. 18, p. 864, September 15, 1912. 6 pp., 5 figs. *d*). Description of a new pump based on the principle of utilizing the external friction between the gas molecules and the walls of a vessel, and the rebound of the molecules at a certain definite angle. In rapidity of operation and degree of exhaust the new pump exceeds the mercury vacuum pump. It can exhaust not only air, but also vapors.

Steam Engineering

ON THE CYCLE OF A STEAM ENGINE (*Sur le cycle de la machine à vapeur*, A. Leduc, *La Revue électrique*, vol. 18, no. 209, p. 195, September 13, 1912. 3 pp., 3 figs. *t*). From a paper read by the author before the Nimes Congress of the French Association for the Advancement of Science. The efficiencies of the Carnot and Rankine cycles are supposed to be very close to each other; in a non-condensing engine working between 200 and 100 deg. cent. (392 and 212 deg. fahr.) they are, but in condensing engines working between 160 and 40 deg. cent. (320 and 104 deg. fahr.) the two efficiencies differ by 13 per cent.

The author proceeds to consider a theoretical case of engine with condenser working at 60 deg. cent (140 deg. fahr.), and using first saturated steam at 200 deg. cent. (392 deg. fahr.), and then steam produced in a boiler at 100 deg. cent. (212 deg. fahr.) and superheated to 200 deg. cent. Let T_0 and T_1 be the temperatures of the isotherms, k real specific heat of water at T_0, L_0 and L_1 latent heats of evaporation at T_0 and T_1, and x quality of steam at the end of adiabatic expansion (on the supposition of the steam being saturated and dry at the beginning). Then

$$\int_{T_0}^{T_1} k\frac{dT}{T}+\frac{L_1}{T_1}=\frac{L_0}{T_0}x \qquad [1]$$

This, calculated by means of the tables of latent heat of Regnault or Henning, with average value $k=1.025$, gives

$$x=0.797 \quad \text{and} \quad \eta=\frac{Q_0}{Q_1}=0.266$$

for the first case.

For the second case let T_1 be the temperature at which the steam is produced (100 deg. cent.), and T_2 (200 deg. cent.) the temperature at

which it is introduced into the cylinder; C specific heat of steam under one atmosphere pressure. Then

$$\int_{T_o}^{T_1} k\,\frac{dT}{T}+\frac{L_1}{T_1}+\int_{T_1}^{T_2} C\frac{dT}{T}=\frac{L_o}{T_o}x \qquad [2]$$

Or, with k=1.01 , and C=0.46,

x=0.996 and η=0.103

or less than half the efficiency of the cycle without superheat. The efficiency of the corresponding Carnot cycle is 0.296; the Rankine cycle is below it by about 10 per cent. and the cycle with superheat by about 65 per cent. The inferiority of the last is explained by an almost total absence of condensation in the cylinder, which results in 0.2×563=112 calories escaping transformation into work. In a practical case, steam is produced saturated at 160 deg. cent. (320 deg. fahr.), and superheated to 300 deg. cent. (572 deg. fahr.), with the condenser at 40 deg. cent. (104 deg. fahr.). Equation [2], with k=1.01 and C=0.50, gives x=0.875, and η =0.265, instead of 0.24, as in the case of saturated steam, thus showing an improvement in efficiency of about 10 per cent. The improvement in the case of the Carnot cycle would be, however, from 0.277 to 0.454, or 64 per cent. The main part of the rest of the paper has been already reported in THE JOURNAL, September 1912, p. 1392, from an article by the author in another periodical.

WATER-TUBE BOILERS WITH AUTOMATIC UNDERFEED FURNACES (*Chaudières aquatubulaires avec foyers automatiques à propulsion inférieure. Le Génie Civil*, vol. 61, no. 21, p. 413, September 21, 1912. 6 pp., 15 figs. and one plate of drawings. *d*). Description of the Sulzer *water-tube boilers and automatic underfeed stokers*, with detail drawings.

ON THE INFLUENCE OF THE MATERIAL AND THICKNESS OF WALLS OF ECONOMIZER PIPES ON THE UTILIZATION OF HEAT (*Über den Einfluss des Materials und der Wandstärke von Ekonomiserrohren auf die Wärmeausnutzung*, Arthur Hanff, *Braunkohle*, vol. 11, no. 25, p. 389, September 20, 1912. 1½ pp. *p*). Discussion of the problem *whether or not wrought-iron pipes* are *preferable to cast-iron ones in economizers* as far as the utilization of heat is concerned; cast-iron pipes being otherwise preferable because they are easier to clean. The coefficient of heat transmission between the gases and water in the economizer is

$$k=\frac{1}{\frac{1}{\alpha_1}+\frac{1}{\alpha_2}+\frac{\delta}{\lambda}}$$

where α_1 is the coefficient of heat transmission between the flue gases and the external wall of the pipe; α_2 same between the inner wall of the pipe and the feedwater; δ thickness of the metal wall in meters; λ coefficient of heat transmission of the wall material. The material and thickness of the wall affect therefore only the single factor $\frac{\delta}{\lambda}$. To show of how little importance this factor is, the author gives the following calculation:

For both cases: a_1=16.142, a_2=660. For cast iron δ=10.5 mm (say 0.4 in.), for wrought iron, 2.5 mm (or 0.1 in); the most advantageous coefficient of heat transmission λ=70 is accepted for wrought iron, and the least advantageous λ=40 for cast iron. Substituting these values in the above formula, the author obtains for wrought iron k=15.755, and for cast iron k=15.70, showing that, even with the above assumptions, cast-iron pipes are practically just as good as wrought-iron pipes as far as transmission of heat is concerned.

EXAMINATION OF THE METHOD OF MR. J.-PAUL CLAYTON FOR THE EXPERIMENTAL STUDY OF RECIPROCATING STEAM ENGINES (*Examen de la méthode de M. J.-Paul Clayton pour l'étude expérimentale de la machine à vapeur à piston*, V. Dwelshauvers-Déry. *Revue de mécanique*, vol. 31, no. 2, p. 105, August 31, 1912. 6 pp., 2 figs. *t*). The author considers Mr. Clayton's method highly valuable, especially for practical purposes. He calls attention to the article which he and A. Duchesne published in the *Revue de mécanique*, July 1901, in which they attempted to show experimentally that the laws of expansion and compression differed from one another. This ought to be expected since during expansion the metallic walls of the cylinder furnish heat to the working fluid, while during compression, or at least a large part of it, it is the superheated fluid that gives up its heat to the surrounding metal.

All the operations and measurements for this article have been made with extreme precision, with the values of all the ordinates and abscissae of the indicator diagrams, and their logarithms given. It is therefore easy to construct a logarithmic diagram. For the expansion the logarithmic diagram *resembles* a straight line, but is not; for the compression the curve is so pronounced, especially towards the end, that there is no possibility of mistaking it. A beginning of a loop appears at the end of compression of the *PV* diagram. Some attributed it to leaks, the author to condensation, and G. Duchesne proved definitely that it was so by compressing in the same cylinder and under the same conditions, incondensible air, and obtaining no loop. *The law of compression of steam in our engines differs therefore from that of expansion.*

As to the practical application of Mr. Clayton's method, the author draws attention to the necessity of taking the greatest precaution for obtaining correct indicator diagrams when they are to be used in this connection. As to the volumes indicated by the *abscissae* of the diagrams: *stroke reducers* with long cord must be absolutely avoided owing to the possibility of whipping: it is essential that when the piston is at exactly 0.1, 0.2, etc. of its stroke, the pencil should be at exactly the same points of *its* stroke, which cannot be done if the stroke reducer consists of a cord which shortens and lengthens, or if the rigid joints have a play. The *clearance* must also be exactly determined either by calculation, or by filling it with water, and still better in both ways. One must not rely too implicitly on the data furnished by the manufacturers as to the ordinates: it is safer to have the instrument calibrated personally. Usually an *average diagram* is made of the head and crank diagram, which is used as if it were a real diagram. This of course is wrong; the

average value of X_e is therefore wrong, although it is very difficult to say just how far it is wrong. The only way to obtain the most exact data possible is to use a surface condenser at each end of the cylinder. The author found in some of his experiments that even with separate valve gears for both sides of the cylinder, and everything, including clearances, on both sides exactly similar, there were still differences between the head and crank diagrams, due probably to different speeds of the piston at both ends of its stroke. The two condensers method is therefore the only way to obtain really exact data.

MANNER OF WORKING OF STEAM GENERATORS (*Innere Arbeitsweise von Dampfgeneratoren*, B. Schapira. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 37, p. 388, September 13, 1912. 2 pp., 9 figs. *gt*). General discussion of the *process of steam generation in a boiler*, mainly with regard to its dependence on the internal circulation.

PURIFICATION OF WATER BY PERMUTIT (*Épuration des eaux par la Permutite*, L. Clera, *Portefeuille économique des machines*, ser. 5, vol. 10, no. 681, p. 130, September 1912. *g*). A general article on the use of permutit, without data of any specific tests.

NEW ARRANGEMENT OF THE SULZER HEATING BOILERS (*Neue Anordnung von Sulzer-Heizkesseln für ausgedehnte Heizanlagen. Dinglers polytechnisches Journal*, vol. 327, no. 37, p. 590, September 14, 1912. 2 pp., 3 figs. *de*). Description and data of tests of the Sulzer *boiler for house heating*. Tests made by the Swiss Association of Steam Boiler Owners have shown an efficiency as high as 86.2 per cent, and an average flue gas temperature, even with the boiler overloaded, below 200 deg. cent. (392 deg. fahr.).

MEANS FOR REMOVING SCALE IN BOILERS (*Über Mittel zur Bekämpfung des Kesselsteines*, E. E. Basch, *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 38, p. 393, September 20, 1912. 2 pp. *p*). Summary of recipes for the prevention of scale in boilers patented at various times in Germany, with short criticisms. Covers practically the same ground as the article reported in THE JOURNAL, February 1912, p. 309.

Testing of Materials and Materials of Construction

EXPERIMENTS ON THE INFLUENCE OF WIDTH IN NOTCH SHOCK TESTS (*Versuche über den Einfluss der Breite bei Kerbschlagproben*, R. Baumann, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 33, p. 1311. August 17, 1912. 4 pp., 10 figs. *e*). The International Association for Testing Materials specified the *width of test bars* for notch shock tests as 30 mm, but for plates decided to consider the thickness of the *plate* as the width of the bar. The width in this case is therefore variable, and its influence on the test results had to be determined. This was partly done by Ehrensberg, and more fully by the author as reported in the present article. The following are the most interesting conclusions arrived at: (*a*) in boiler plates after annealing the work required for rupture has decreased 116 per cent in case of "broad" plates, and increased 11 per cent with "narrow" plates; (*b*) in the case of square iron 40 mm on a side the work required for rupture suddenly decreased when the width

passed 1.5 cm. The material which appeared as tough when not broader than 1.5 cm, acted as brittle in broader test pieces. The author explains it by the supposition that a wide piece does not permit the contraction of area and flow of material, and that the change of state occurring in broad pieces is entirely different from that in narrow.

The author thinks therefore that the specifications of the International Association for Testing Materials ought to be revised, and a narrower bar specified. The article contains full data of the tests, as well as data of tests on cast iron.

USEFUL EXOTIC WOODS AND THEIR APPLICATION FOR TECHNICAL PURPOSES (*Exotische Nutzhölzer und ihre Verwendung in der Technik, Th. Wolff. Prometheus*, vol. 23, nos. 1191 to 1193, pp. 742, 759 and 774, August 24, 31 and September 7, 1912. 12 pp. *dp*). Discussion of the application for technical purposes and description of properties of exotic, i.e. *non-european, woods*, viz. quebracho (Argentinian wood, extremely hard, unaffected by air or water, good for railway ties, telegraph poles and bridges); teak; pock (has the greatest specific weight, extremely hard); hickory; pitch-pine; redwood; eucalypt; Australian cypress and blackwood; mahogany; cedar and cypress. The article contains only general information without test data.

ACTION OF ELECTRICITY ON REINFORCED CONCRETE (*Les effects de l'électricité sur le béton armé*, F. H. *Le Génie Civil*, vol. 61, no. 20, p. 396, September 14, 1912. 4 pp., 23 figs. *e*). Description and data of tests on the *action of electricity (stray direct currents) on reinforced concrete* made at the Technical School at Darmstadt, Germany, and University of Vermont in this country. Only the former will be reported here; for the latter see *Engineering News*, July 1911. With concrete blocks placed in air on a layer of moist sand, or in sweet water, and subjected to the passage of a current of 0.1 ampere at 140 volts, fissures appeared after a period of from 50 to 110 days, the location of the fissures depending entirely on the position of the anode; in blocks placed in salt water they appeared at the end of from 20 to 70 days, and the concrete was colored in the neighborhood of the fissures first greenish gray (ferrous oxide), and after some exposure in the air, brown (ferric oxide). Fissures appeared in blocks immersed in chalk water in about the same time. A series of experiments led to the belief that these fissures were produced not by an evolution of gases, but by a thickening of the iron anode enclosed in the concrete, owing to superficial oxidation. By further tests it was found that the resistance of the reinforcing members to being torn out is increased by the action of the current, which may be explained by the formation of a layer of oxide adhering to the concrete more strongly than the original clean iron.

CONTRIBUTION TO THE PROBLEM OF MECHANICAL TESTING OF SOFT RUBBER (*Beiträge zur Frage der mechanischen Weichgummiprüfung*, K. Memmler, *Mitteilungen aus dem Kgl. Materialprüfungsamt zu Gross-Lichterfelde West*, vol. 30, no. 3, p. 139, 1912. 11 pp., 10 figs. *e*). Description and data of tests to determine the *resistance of soft rubber to compression stresses*

and wear, such as it is subject to when used in automobile tires. For previous tests of tensile strength of rubber by the same institution see THE JOURNAL, March 1912, p. 430.

EXPERIMENTS ON REDUCTION OF STRESS BY ROUNDING OFF SHARP ANGLES (*Versuche über die Spannungsverminderung durch die Ausrundung scharfer Ecken*, E. Preuss, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no 34, p. 1349, August 24, 1912. 5 pp., 28 figs. *e*). Description of and data from experiments made at the testing laboratory of the Darmstadt Technical High School on the *reduction of stresses* in metal members *by rounding off sharp angles*. The author states that although the fact that the stresses are particularly large at sharp angles was known for a long time, there are practically no experiments available showing numerically the precise influence of the radius of curvature on the stress at and near the point of flexure. Fig. 6 shows graphically how the change in the radius of curvature affects the distribution of lines of equal stresses. Among other things, these experiments have established the fact that the

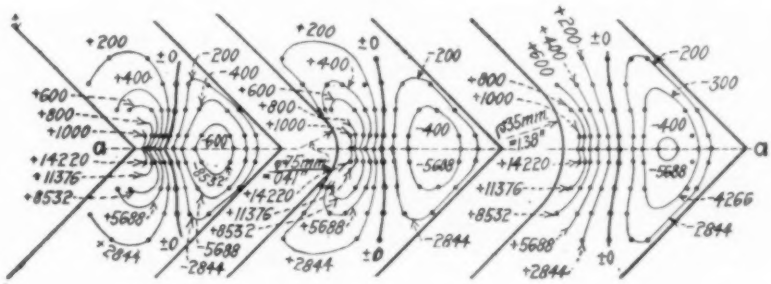

FIG. 6 DISTRIBUTION OF LINES OF EQUAL STRESSES FOR VARIOUS RADII OF CURVATURE
+Tensile Stress; — Compression; 0 Points constructed from Measurements: above Line *aa* in kg/qcm, below in Lb. per Sq. In.

neutral point of stresses as found from actual measurements does not coincide with that determined theoretically. A complete account of these experiments will appear shortly in the *Mitteilungen über Forschungsarbeiten*.

USE OF TRASS IN REINFORCED CONCRETE (*Gebruik van tras bij gewapend beton*, E. Jacobs. *De Ingenieur*, vol. 27, no. 26, p. 734, September 7, 1912. 4 pp., 4 figs. *eA*). Data of tests for determining the influence of the proportion of trass in reinforced concrete on the properties of the latter. The tests were made with "Heidelberg" portland cement and Andernach trass to supplement the tests of Professor Van der Kloes (presented at the Copenhagen congress of the International Association for Testing Materials). The author concluded that in beams with maximum 2 per cent reinforcement (which is seldom exceeded) concrete with say ¾ portland cement, ¼ trass, 2 sand, and 2 gravel is in general just as strong as concrete with pure cement. The article gives full data of the tests, and drawings of stressed sections of the beams tested.

DETERMINATION OF THE PROPORTION OF ORIGINAL INGREDIENTS IN CEMENT, MORTAR AND CONCRETE AFTER THEIR SETTING (*Die Bestimmung des Mischungsverhältnisses von abgebundenem (erhärtetem) Zementmörtel und -beton*, H. Burchartz. *Mitteilungen aus dem Kgl. Materialprüfungsamt zu Gross-Lichterfelde West*, vol. 30, no. 3, p. 117, 1912. 4½ pp. *ep*). It is sometimes necessary to determine the proportion of original ingredients in mortar or concrete after their setting, e.g. in case of an accident to a building when it is desired to find out whether or not it was due to poor mixture. The present article indicates the conditions when such a determination can be made, and the methods of procedure.

TESTS OF SLAG CEMENT FOR THE PURPOSE OF COMPARING IT WITH PORTLAND CEMENT (*Prüfung von Eisenportlandzement im Vergleich zu Portlandzement*, M. Gary and H. Burchartz. *Mitteilungen aus dem Kgl. Materialprüfungsamt zu Gross-Lichterfelde West*, vol. 30, no. 3, p. 122, 1912. 8 pp. *ce*.) Data of supplementary tests for the determination of the properties of *slag cement* as compared with portland cement. For the main investigation see THE JOURNAL, May 1912, p. 805.

ACTION OF OIL CONCRETE AND REINFORCED CONCRETE (*Einwirkung von Oel auf Beton oder Eisenbeton. Deutsche Bauzeitung*, no. 11, 1912, through *Metall-Technik*, vol. 38, no. 35, p. 288, August 31, 1912. ½ p. *p*). From a symposium of answers to a general inquiry sent out by the German Cement Association to its members. The action of *fat oils*, i.e. animal or vegetable, is different from that of mineral oils. The former become "rancid," i.e. decompose into glycerin and acid of fat, the latter entering into chemical composition with the calcium of the cement and thus decomposing it, which is especially noticeable in the case of porous mortars. In practice, machinery foundations subject to the action of constantly dropping lubricating oils and grease, are gradually affected and destroyed. Cement tanks for oil are sometimes made useless after a service of a couple of months. On the other hand *mineral oils* form no acids having affinity to calcium, and therefore do not affect cement at all.

PREVIOUS TREATMENT OF SHEETS BEFORE THEIR USE IN BOILER SHOPS (*Traitement préalable des tôles avant leur emploi en chadronnerie*, Desjuzeur. *La Houille blanche*, vol. 11, no. 7, p. 178, July 1912. 8 pp., 5 figs. *epA*). Investigation of the problem of *rendering boiler sheets* by a previous treatment *less sensible to irregularities of working* during the manufacture of boilers. The author does not reach any final conclusions, but shows that boiler sheets hardened, tempered and then cold worked, are in about the same condition as sheets first worked and then tempered; this may be of some value because it is probably easier to harden and temper the sheets first rather than temper them after cold working, although the cost of hardening must also be considered. The second part of the article contains data of a series of interesting tests the purpose of which was to determine whether *punching of rivet holes* in boiler plates produces only a permanent deformation in the neighborhood of the hole, or starts cracks as well, and whether this may be best remedied by annealing the boiler plate, or by reaming out the punch hole, and in the latter case, how much must be reamed out. Tests were made with plates

of various thicknesses, and it was found (full data are given in the article) that plates 10 mm (say 0.4 in.) thick not annealed after punching cracked after bending through 53 to 59 deg., while the same plates punched and subsequently annealed could be bent through 110 deg. without cracking; previous hardening and annealing makes it comparatively safe to bend plates with punched and reamed out holes, but does not permit of bending plates with holes simply punched, and cannot therefore be substituted for reaming. Plates 15 mm (say 0.6 in.) thick, when simply punched, cracked after bending through 107 deg., and broke when bent to 131 and 147 deg., but plates with drilled, or punched and reamed, holes (even reamed only to 2 mm, or 0.078 in.) could be bent with only slight flaws at the edges. Annealing had the same effect as reaming, in permitting plates to be bent without cracks. In general it appears that reaming of only 2 mm (0.078 in.) in diameter removes the part of the metal altered by punching, and that annealing is also effective in restoring to the metal its original ductility, thus proving that punching produces a permanent deformation of the metal, and not cracks, since in the latter case annealing would do no good. But though more rapid and probably less expensive than reaming, annealing has the disadvantage of tending to change the shape of the plate, especially if it is thin, and to increase the non-coincidence of holes in the assembled boiler shell; besides, it does not remove the burr of the punch which is in the way of a good contact at riveting. Reaming seems to be better in every way, but if only of 2 mm (0.078 in.) in diameter, though sufficient according to the author's tests to remove the permanently set metal, may not be large enough in the case of superimposed plates, owing to a possible non-coincidence of the punched holes; since the divergence of the holes may be normally supposed to amount to about 1 mm (0.039 in.), the hole ought to be reamed to 4 mm (0.16 in.) so as to be sure to take care of the eccentricity of the hole in the lower plate.

Finally the author reports some interesting expanding tests on boiler sheets, which though not directly connected with the above tests, prove the importance of reaming holes after punching. The tests were made on a plate from a shell of a water conduit under pressure which was broken in an accident to the conduit. Rings were cut out from the plate, 100 mm (3.9 in.) in diameter, with the hole in the center and submitted to the expanding test. The best showing a ring with a punched hole made was when a ring with a hole 22.5 mm in diameter broke after the hole was expanded to 30 mm, while rings with drilled holes made from the same sheet broke only after a hole 22 mm in diameter was expanded to 55 and even 69 mm in diameter, thus showing that a plate with a drilled hole is several times stronger than one with a punched hole.

TESTING OF MATERIALS IN A MODERN AUTOMOBILE FACTORY (*Materialprüfung in einer modernen Automobilfabrik, E. Renisch. Auto-Technik*, vol. 1, nos. 15, 16, 17 appended to *Allgemeine Automobil-Zeitung*, vol. 3, nos. 33, 35, 37. 8 pp., 46 figs. *dp*). Description of the testing laboratory of the Protos Automobile Company, Nonnendamm, Germany. In addition to the usual apparatus for mechanical testing, this laboratory is fitted

with a complete installation for metallographic investigation and micro-photography. In testing, particular attention is paid to the appearance of the surface of fracture, as well as to tests for homogeneity of structure of the metal in general, and in particular to the presence of segregated particles of sulphur and phosphorus. For the latter purpose the Heyn test is used: the piece of metal is ground and polished along and across the grain, and etched by suitable chemicals. For etching of a polished surface, see A. Martens and E. Heyn, Handbuch der Materialienkunde (Berlin 1912), vol. 2, pt. A, p. 178 ff.

Pure Zinc Pipes (*Rein-Zinkrohre, M. Haustechnische Rundschau*, vol. 17, no. 5, p. 51, September 1, 1912. 1 p., 15 figs. *d*). Description of *zinc*

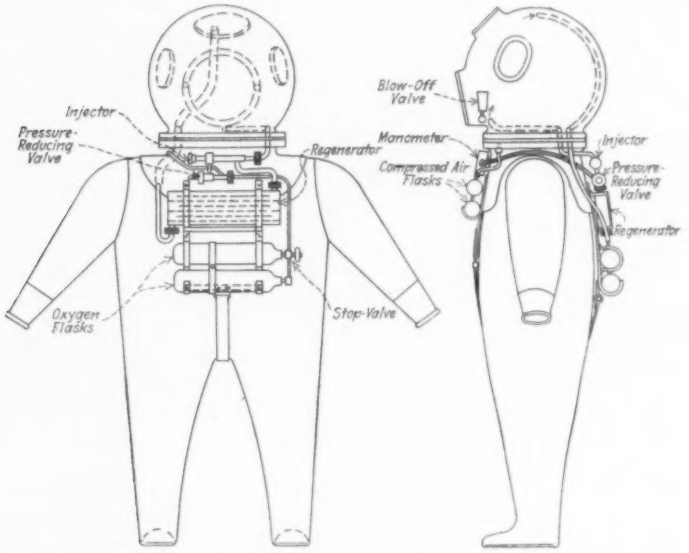

Fig. 7 Diving Apparatus Westfalia

pipes of German manufacture for use in *house water piping*. The advantages claimed for them in comparison with lead and galvanized iron pipes are: they are less affected by water and its impurities than either of the latter; they do not give rise to local electric currents due to the contact of two different metals (in the case of galvanized iron pipes) and rapidly destroying the pipes; even if some zinc be dissolved by the water, it is harmless. Tests at the Royal Laboratory at Gross Lichterfelde West have shown that mechanically zinc pipe is much stronger than lead pipe. It also gives a very strong welded joint.

Miscellanea

New Diving Apparatus Westfalia (*Die neuen Tauchgeräthe "Westfalia."* Grahn. *Glückauf*, vol. 48, no. 36, p. 1448, September 7, 1912. 3 pp., 5 figs. *d*). By invitation of the German Navy Office several firms

have endeavored to construct a *diving apparatus without air pipe*, such as might be used by the crew of a submarine in case of accident. Such an apparatus built by the Lubeck Dredging Machinery Company was described in THE JOURNAL, September 1912, p. 1584. The present article treats of one by the Westfalia Armaturen- und Maschinenfabrik in Gelsenkirchen, Germany. Both factories had, previous to the construction of diving apparatus, a large experience in the manufacture of similar apparatus for miners and firemen. As shown in Fig. 7 the diver carries on his back a regenerator tank taking care of carbon dioxide and water vapor from breathing, while below it are steel flasks with gas for breathing. This gas is not pure oxygen, which is poisonous when at high pressure (cp. *Deutsche medizinische Wochenschrift*, no. 32, 1912), but

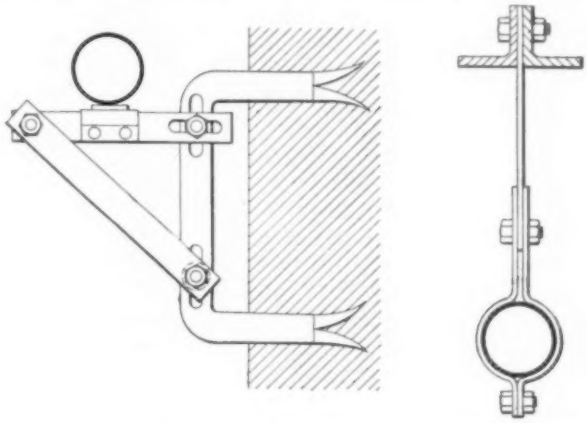

FIG. 8 FRANZ MAASS AND HARDT ADJUSTABLE PIPE SUPPORT

a mixture of oxygen and compressed air in such proportion that even at a depth of 50 m (164 ft.) corresponding to a pressure of 6 atmospheres absolute the contents of oxygen does not exceed 23 per cent. Various ratios of the mixture are used for work at various depths; thus, for 20 to 30 m (65 to 98 ft.) 45 per cent oxygen and 55 per cent nitrogen, while for depths from 50 to 60 m (164 to 196 ft.) 30 per cent oxygen and 70 per cent nitrogen. The oxygen and air or nitrogen are not kept in separate vessels, but the flask is first filled with compressed air up to a certain pressure, and the oxygen is then pumped in, special tables being used for the determination of the pressure corresponding to certain percentages of oxygen in the mixture. The pressure-reducing valve between the steel flasks and injector is so arranged as to vary automatically the amount of gas admitted in proportion to the water pressure, the diving outfit being thus always filled and permitting of easy and comfortable breathing. The compressed gas mixture serves also to drive the injector which, with the help of suitably arranged piping, takes the vitiated air from the upper part of the helm and returns it thither after it has passed through the regenerator and become free of the carbon dioxide and mois-

ture. On the front side are placed two flasks with compressed air, to be used for rapidly filling the diving suit when the diver wants to come to the surface quickly. The manometer permits him to see what supply of air and oxygen he still has at his command. The article also describes a second diving outfit without helm or diving suit, for use for short stay under water or in case of emergency, e.g. accident to a submarine.

LIQUEFACTION OF HYDROGEN WITH NO COLD LOSSES (*Verflüssigung des Wasserstoffs bei Vermeidung von Kälteverlusten*, K. Olszewski, *Krak. Anz.*, 1912. pp. 1 to 27, through *Beiblätter zu den Annalen der Physik*, vol. 36, no. 18, p. 969. *d*). Description of the Improved *Olszewski hydrogen liquefier, thermostat* for temperatures from -190 to 0.5 deg. cent., and gas liquefaction plant in Krakau, Austria.

POROUS METALS (*Les métaux porcaux*, H. I. Hannover, *Revue de métallurgie*, vol. 9, no. 9, p. 641, September 1912. 5 pp., 7 figs. *dt*). General remarks on the solidification of metallic alloys, and description of the author's method of producing porous metals, chiefly lead plates for accumulators (eutectic alloys).

ADJUSTABLE PIPE SUPPORTS (*Verstellbare Rohrträger*. *Prometheus*, vol. 23, no. 1194, p. 799, September 14, 1912. 1 p., 5 figs. *d*). Description of an *adjustable pipe support* to be used either when the position of a pipe has to be changed, or when the original support has been placed incorrectly. By unscrewing the two bolts, the pipe may be adjusted for practically any desired position within a fairly wide range of variation (Fig.8).

DREDGES (*Les dragues*, A. Baril, *Revue de mécanique*, vol. 31, no. 2, p. 111, August 31, 1912. 43 pp., 82 figs. *d*). Description of dredges, mainly for harbor work, European types.

Supplementary References

PREHEATING THE AIR IN INTERNAL COMBUSTION ENGINES (April and May, 1912, pp. 615 and 793). Cp. another report of same article in *Power*, October 15, 1912, p. 565, and an article on the same subject in *Gas Engine* (Preheating Gas Engine Charge, no. 10, 1912, p. 538).

THEORY OF STEEL-WELDING (June 1912, p. 961). The same article reported in *Acetylene Lighting and Welding Journal*, September 1912, p. 205.

SOMETHING ABOUT DIVING (October 1912, p. 1584). For a description of the Lubeck diving apparatus see also *Scientific American*, October 5, 1912, p. 284.

REPORTS OF MEETINGS

MEETINGS ON SMOKE ABATEMENT

The problem of Smoke Abatement has been considered at two recent meetings of the Society, George H. Perkins, head of the engineering department of the Lowell Textile School, giving his report on the International Smoke Abatement Exhibition and Conference, held in London in March and April 1912, at which he represented the Lowell Textile School and also the Society.

At the first of these, a joint meeting of the Philadelphia Section and the Engineers Club of Philadelphia, held in that city on October 5, a paper on Methods and Means of Smoke Abatement, was presented by the author, O. R. McBride of Pittsburgh, following Mr. Perkins' report, and was profusely illustrated with lantern slides. Talks on the sanitary aspect of the smoke problem were given by Dr. Andrews Allen, representing the County Medical Association, and Dr. Anders, both tracing the relation between the presence of smoke in the atmosphere and the spread and virulence of disease. John M. Lukens, chief of the Bureau of Boiler Inspection of Philadelphia, gave a brief sketch of the municipal efforts in the direction of smoke abatement in that city, and quoted figures to show that about 500 smoke-consuming appliances have been installed there. Henry S. Morris recommended for this purpose the wider use of gaseous fuel as both a cleaner and on the whole a more economical method than burning coal in the furnace. Henry J. Hartley, on the other hand, ascribed a large part of the smoke nuisance to the desire of manufacturers to get out of a boiler more than its capacity, and to their trying to force through more fuel than the grates can take care of properly. W. C. Furber said that smoke inspectors are on duty in Washington until six o'clock each evening and that the hotels may smoke as much as they want to after that hour. Prof. Robert H. Fernald insisted on the necessity of eliminating the political factor from the problem of smoke prevention, and of exceedingly careful study of the type of furnace and other conditions. In conclusion he made favorable mention of the work done in this direction in Cleveland.

At a meeting of the members in New York and vicinity, held on October 8, more than 300 members and representatives of various municipal and technical organizations were present. Following Mr. Perkins' report, Paul P. Bird told how the smoke problem is being solved in Chicago and what the smoke department there is doing in this respect. Prof. O. P. Hood, representing the United States Bureau of Mines, followed him with a statement of the attitude of the Government on this problem. Lieut. James Reed, formerly assistant director of Public Works of Philadelphia, and John M. Lukens, pointed out the important role of the railroads in this respect.

The scientific side of the problem of smoke prevention was taken up by Prof.

L. P. Breckenridge of Yale University, who concluded by stating that with the data now available, and knowing the volatile content in a coal and rate of combustion, it is entirely possible to design furnaces properly. The same line of discussion was continued by Professor Kent and Dr. Jacobus, who showed that smoke may be prevented and explained the difficulties in the way of its complete elimination. Mr. Adler closed the discussion by a review of the principles of combustion, demonstrating that the essential conditions of smoke abatement are a long flame length, more refractory and better firebrick, and a high chimney.

Written discussions were submitted by Charles T. Main, Horace C. Porter of the Bureau of Mines, Pittsburgh, Prof. J. A. Moyer, Perry Barker, H. V. O. Coes, Dean C. H. Benjamin, E. C. Fisher, C. S. Davis of the William B. Pierce Co., Buffalo, N. Y., George R. Henderson, J. S. Shoemaker of Cumberland Mills, Me., E. A. Uehlung, and M. C. Huyette of Murphy Iron Works, Buffalo, N. Y.

BOSTON MEETING, OCTOBER 16

The first meeting of the season in Boston was held on October 16, the Boston Society of Civil Engineers and the Boston section of the American Institute of Electrical Engineers coöperating with the Society. A paper on the Water Power Development of the Mississippi River Power Company at Keokuk, Iowa, was presented by Hugh L. Cooper, chief engineer of the company. By way of introduction Mr. Cooper showed a few slides illustrating notable features of other undertakings. The immense undertaking at Keokuk and its several parts in various stages of progress were very fully illustrated and described.

In the discussion of the paper which followed, E. L. Galusha, engineer with the Stone and Webster Engineering Corporation of Boston, presented an illustrated description of the main electrical features of the plant, including the generators, switching and other power-house equipment, and transmission line construction.